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VIBRATION CHARACTERISTICS OBTAINED
FROM A 1/5-SCALE MODEL AND
FROM A FULL-SCALE SATURN SA-1

by John S. Mixson and John J. Catherine

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Langley Station, Hampton, Va.



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SUMMARY

Design of the structure and control system of a large, flexible, liquid-fueled booster vehicle requires knowledge of its vibration characteristics because of the possibility that vibrations can cause critical stresses in the structure or instabilities in the control system. In order to explore the feasibility of using reduced-scale models in place of the usual full-scale vehicle to obtain the necessary resonant frequencies, mode shapes and damping, a 1/5-scale model of the Saturn SA-1 has been built and vibration tested at Langley Research Center. The results of these vibration studies of the 1/5-scale Saturn model are compared herein with results of a ground vibration survey of a full-scale Saturn vehicle performed at Marshall Space Flight Center.

A brief description is given of the full-scale Saturn vehicle, the 1/5-scale Saturn model, and the suspension systems used to support them for the vibration tests. The scaling concepts employed in the design and testing of the model are discussed. Comparisons of the model and full-scale vibration test results show that the model and full-scale first-bending-mode frequency parameters (obtained by multiplying the frequency in cycles per second by the vehicle length) are in agreement when the rigid body rocking frequency parameters agree, that the model and full-scale damping factors are approximately the same for most modes, and that the first three mode shapes are in qualitative agreement. The model and full-scale vehicles both exhibit a nonlinearity of the first bending mode response, which is characterized by a decrease in resonant frequency with an increase in vibration amplitude. The change of first-mode frequency of the model due to nonlinearity was about the same magnitude as the change of frequency due to suspension-system stiffness changes.

INTRODUCTION

A ground vibration survey has become a normal part of the development program of large launch vehicles. The resonant frequencies, mode shapes, and damping of the vehicle elastic modes are the principal result of such a survey and are used to verify the design of the vehicle control system and structure. As vehicle size has increased, however, the cost, effort, and time required for

ground vibration testing has also increased. In addition, it has become very difficult to design a suspension system which will accurately simulate the desired free-free vehicle end conditions.

In the search for less expensive and easier methods of obtaining the necessary vibration data, the consideration of dynamically scaled models is natural, in view of the valuable contributions of model studies in the field of airplane dynamics. In order to explore the feasibility of obtaining and using such model results, as well as to study the vibrations of a clustered tank configuration, a 1/5-scale model of the Saturn SA-1 launch vehicle has been constructed and its vibration characteristics determined at Langley Research Center. This model program was initiated and carried out in cooperation with the Marshall Space Flight Center, where the full-scale Saturn SA-1 vibration programs were performed. This model was designed and constructed to duplicate as nearly as possible all important structural characteristics of the full-scale vehicle.

The purpose of this report is to present comparisons of the bending vibration characteristics of the 1/5-scale Saturn model with those of the full-scale Saturn SA-1 vehicle. The resonant frequencies, mode shapes, and damping of the elastic bending modes of the 1/5-scale Saturn model have been determined for a range of weight conditions and with two different suspension systems. These results, presented in references 1 and 2, show the unusual frequency spectrum and mode shapes of the Saturn vehicle associated with the clustered arrangement of the booster tanks. A description of the full-scale Saturn SA-1 vibration test vehicle, of the suspension system used to support the vehicle for the vibration tests, and the resulting resonant frequencies, mode shapes, and damping are presented in reference 3.

During the model and full-scale test programs Mr. Emil A. Hellebrand of Marshall Space Flight Center cooperated in the planning of the model test program with the needs of the full-scale program in mind and in effecting prompt exchange of test results between the model and full-scale programs. In a members report to the NASA Research Advisory Committee on Space Vehicles he compared the vibration test results and drew the conclusions that the model test results gave a good prediction of the resonant frequencies, damping, and unusual mode shapes of the complex structure of the Saturn and were therefore helpful in preparing and refining the full-scale vibration test program.

In the present paper the model and full-scale vibration test vehicles and test setups, including the suspension systems used to support the vehicles, are described, and the scaling concepts employed in the design of the model are discussed. The resonant frequencies, and the associated mode shapes and damping of the model and full-scale vehicles are compared, and a nonlinear variation of response with shaker force is discussed.

Unpublished data obtained from D. G. Douglas of the Marshall Space Flight Center on the linearity of the response of the full-scale Saturn vehicle are also presented.

SYMBOLS

E	Young's modulus, lb/sq in.
f	frequency, cps
\bar{f}	frequency parameter, fL, ft-cps
G	acceleration of gravity
\bar{G}	acceleration, G units
g	damping factor, $\frac{\log_e \frac{x_0}{x_n}}{2\pi n}$
I	area moment of inertia, in. ⁴
I'	mass moment of inertia, lb-in-sec ²
L	vehicle length, in. or ft
m	mass, lb-sec ² /in.
n	number of cycles of vibration used in determining damping
x_0	initial vibration amplitude, in.
x_n	vibration amplitude after n cycles, in.
\bar{x}	vibration amplitude of vehicle nose divided by L
μ	Poisson's ratio
ρ	mass density, lb-sec ² /in. ⁴
Subscripts:	
F	full scale
M	model

SATURN DESCRIPTION

Flight Vehicle

The Saturn configuration of interest in this report is designated the "Saturn I Block I"; the term "Saturn SA-1" refers to the first flight vehicle

of this configuration. During flights of this vehicle the first, or booster, stage is the only live stage; the upper stages are dummies designed to have the same overall weight, center of gravity, moment of inertia, and aerodynamic shape as future "live" upper stages planned for use with this booster. The complete vehicle is 162 feet high and weighs about 900,000 pounds at lift-off. Eight liquid-fuel rocket engines produce a total of about 1.3 million pounds of thrust. The general characteristics of the Saturn I Block I configuration are discussed in reference 4 along with the characteristics of other Saturn launch vehicle configurations.

The principal structural features of the Saturn SA-1 vehicle are illustrated in figure 1. (Photographs of the full-scale booster structure during assembly are presented in refs. 5 to 7.) The booster stage consists of nine tanks: eight 70-inch-diameter outer tanks clustered around a 105-inch-diameter center tank. The tanks are connected at the lower end by the thrust structure and at the upper end by the spider beam. The center tank and four alternating outer tanks carry liquid oxygen (lox) and the remaining four outer tanks carry fuel. The center tank is firmly attached to the corrugated barrel at the lower end and to the spider beam at the upper end. This structure is the principal load-carrying structure of the booster and normally takes 60 percent of the thrust loads from the engines. The outer tanks are attached to the outriggers by means of two ball-and-socket type joints at the lower end of each tank. The type of joints used are illustrated in figure 2. The outer lox tank upper joints (fig. 2) will transmit longitudinal loads so that these outer lox tanks carry the remaining 40 percent of the engine thrust loads. The fuel-tank upper joints, shown also in figure 2, cannot transmit longitudinal loads; therefore, the fuel tanks carry only their own weight, the fuel weight, aerodynamic loads, and some side loads due to vehicle bending. With this type of attachment the outer tanks are not restrained from motion relative to the center tank or relative to each other.

The second stage of the Block I Saturn vehicle illustrated in figure 1 consists of an outer shell which is connected by means of eight radial truss assemblies to an inner water ballast tank. The outer shell is the principal structural member of the second stage; it supports the weight of the ballast tank, which constitutes 70 percent of the second-stage weight when water filled, and the weight of the third stage. The third stage is simply a water ballast tank which also supports the weight of the dummy payload.

Full-Scale Dynamic Test Vehicle

The Saturn I Block I dynamic test vehicle, designated SA-D1, is shown in the dynamic test tower at Marshall Space Flight Center in figure 3. This vehicle is virtually the same, structurally, as the flight vehicle. In the vibration tests, however, the liquid oxygen and fuel were replaced by de-ionized water. As shown in reference 3, the total weight of SA-D1 was within 3 percent of the flight vehicle weight. The experimental test program and the resulting measured vibration characteristics of this vehicle are described in reference 3.

The suspension system used to support SA-D1 for the ground vibration tests is illustrated in figure 4. The vehicle was suspended from the support ring of

the vibration tower by cables arranged in series with springs to provide flexibility and attached to each of the eight outriggers. The leveling device used on SA-D1 was a hydraulic cylinder, which was also used to raise the vehicle off the outrigger holddown structure. (When testing was not in progress, the outriggers were fastened down directly to the base of the tower structure.) A typical bank of springs and a hydraulic cylinder are shown in figure 5, and a lower connecting link is shown in figure 6. In order to keep the rigid-body rocking frequency as low as possible and to investigate the effects of suspension-system variations, this suspension system was "softened" by removing springs from the spring banks, by removing some of the cables (that is, supporting the vehicle from fewer than eight outriggers), or by a combination of these. Details of the suspension configuration for each weight condition are given in reference 3.

1/5-Scale Saturn Model

Scaling.— The 1/5-scale model of the Saturn SA-1 launch vehicle was constructed for the purpose of investigating vibration characteristics; therefore, the important parameters to be considered in scaling were the mass and stiffness magnitudes and distributions. The type of scaling chosen was a uniform reduction of dimensions to one-fifth of the full-scale values wherever possible. This type of scaling was chosen because of the structural complexity of the Saturn launch vehicle with the resulting difficulty of determining accurate equivalent stiffness and mass properties for the many complex multiple-beam trusswork assemblies incorporated in the vehicle. Scaling of linear dimensions determined the scale factor of one-fifth from practical considerations of the minimum material thickness required for satisfactory fabrication. Careful attention was given to accurate reproduction of the important structural members and the fittings joining the components together and to the use on the model of the same materials as those used on the full-scale vehicle, so that the model stiffness would represent as nearly as possible, on a reduced scale, the stiffness of the full-scale vehicle. To represent the full-scale mass distribution accurately, ballast weights were substituted for the nonstructural components such as aerodynamic fairings and fuel piping which were omitted from the model.

From these considerations the following scale factors were determined:

Selected values:

Typical lengths:

$$\frac{L_M}{L_F} = \frac{1}{5}$$

Material properties:

$$\frac{E_M}{E_F} = \frac{\rho_M}{\rho_F} = \frac{\mu_M}{\mu_F} = 1$$

Computed values:

Mass:

$$\frac{m_M}{m_F} = \frac{\rho_M}{\rho_F} \left(\frac{L_M}{L_F} \right)^3 = \frac{1}{5^3}$$

Mass moment of inertia:

$$\frac{I'_M}{I'_F} = \frac{\rho_M}{\rho_F} \left(\frac{L_M}{L_F} \right)^5 = \frac{1}{5^5}$$

Cross-section moment of inertia:

$$\frac{I_M}{I_F} = \left(\frac{L_M}{L_F} \right)^4 = \frac{1}{5^4}$$

Bending frequency:

$$\left(\frac{f_M}{f_F} \right)_{\text{bending}} = \left[\frac{EI_M}{EI_F} \frac{m_F}{m_M} \left(\frac{L_F}{L_M} \right)^3 \right]^{1/2} = \frac{L_F}{L_M} = 5$$

Sloshing frequency:

$$\left(\frac{f_M}{f_F} \right)_{\text{sloshing}} = \left(\frac{\bar{G}_M}{\bar{G}_F} \frac{L_F}{L_M} \right)^{1/2} = 5^{1/2}$$

where the subscripts F and M indicate full-scale and model values, respectively.

Since the bending frequencies scale directly with the scale factor, whereas the sloshing frequencies scale with the square root of the scale factor, the bending-sloshing frequency relationship of the full-scale vibration test is not maintained on the model. Thus, the interaction of vehicle bending with sloshing on the model will not represent directly the full-scale situation. For a configuration such as Saturn where the first sloshing frequency is smaller than the first bending frequency, the reduction to model size separates the frequencies relative to the ground vibration vehicle. This reduction tends to uncouple further the sloshing and bending modes, which are already uncoupled in the ground vibration tests as compared with the flight vehicle which operates at accelerations up to several times that due to gravity.

Description.- The 1/5-scale dynamic model of the Saturn SA-1 vehicle is shown in the vibration testing tower at Langley Research Center in figure 7. The model length is 32.4 feet and the full-scale length is 162 feet. The

overall weight of the 1/5-scale Saturn model is compared on a scaled basis with the weight of the full-scale dynamic test vehicle in figure 8. For these comparisons the overall weight was divided by the cube of the length of the respective vehicles and the center-of-gravity distances from the base were divided by the vehicle length. Figure 8 shows the model and full-scale weights and center-of-gravity location to be in good agreement.

The important structural characteristics of the Saturn vehicle illustrated in figure 1 were reproduced on the model. For example, figure 9 is a photograph of the booster structure of the model during assembly and illustrates the details used in the model construction. The outer-tank attachment joints shown in figure 2 were also carefully duplicated; a photograph of the model outer-tank upper-attachment joints is shown in figure 10. In figure 11 photographs of the model and full-scale outrigger structure are presented. Figure 11 shows that structural details such as skin corrugation, built-up riveted beams, tension rods, and longitudinal stiffeners in the outer tanks have been duplicated on the model; however, nonstructural components such as aerodynamic fairings and piping supports have been omitted. The number of rivets was not duplicated on the model; however, good riveted-construction shop practice was followed and convenient sized rivets were used and, where possible, the total rivet shear area of the model was approximately 1/25 of the full-scale rivet shear area.

Two structural simplifications, which were later found to be significant, were made on the model: (1) the engine support and construction were simplified, and (2) some ring frames were omitted from the outer shell of the second stage of the model. A closeup view of the Saturn model showing the simulated engines is presented in figure 12. The simulated engines were designed to have the proper weight, center-of-gravity location, and moment-of-inertia values scaled down from full-scale values. The engines were then attached to the thrust pads by a screw-type joint rather than by a simulated gimbal and actuator system. Even though details of the engine vibratory motion were not expected to represent full-scale motions because of this simplification, it was felt that the overall effects of the engines on the model elastic modes should represent the overall effects of the full-scale engines.

The other simplification is shown in the end view of the second stage of the 1/5-scale Saturn model in figure 13. Figure 13 shows the outer shell, the eight radial truss assemblies, and the inner water ballast tank. Three ring frames were used to stiffen the outer shell. In the model design this use of three ring frames in place of the seven that appear in the full-scale vehicle was not thought to be critical to the model bending modes. The test results to be discussed subsequently indicated, however, that some of the higher frequency bending modes of the model were affected by this reduction in the number of ring frames.

Suspension systems.- Two suspension systems were used for two series of vibration studies of the 1/5-scale Saturn model. The first series of studies was performed using the suspension system illustrated in figure 14. This suspension was developed at Langley Research Center and its characteristics are discussed in detail in reference 8. As illustrated in figure 14, the weight of the model is supported by the two vertical cables attached to the support bar at the base of the model. (The attachment of the support bar to the outriggers of

the 1/5-scale model is shown in fig. 15.) The model is prevented from toppling over by the horizontal restraining cables connecting the top of the model to the vertical support cables. The rigid-body-rocking suspension-system frequency can be varied by changing the distance between the vertical support cables at the top where they are attached to the vibration tower structure, and thus changing the tension in the horizontal restraining cables. The shakers are attached, as shown in figure 15, so that vibration occurs perpendicular to the plane of the two vertical support cables. This support system holds the model in a vertical position but for small vibratory deflections it exerts very small restraining forces in the plane of vibration. The vibration characteristics of the Saturn model with this suspension system are reported in detail in reference 1.

The second series of studies was performed with an eight-cable suspension system designed to simulate the suspension used on the full-scale vibration test vehicle. The term "eight cable" is used as a convenient name for this system, the principal feature of which is the support of the model by cables attached to the ends of the outrigger. This model suspension system had the same general configuration as the full-scale suspension illustrated in figure 4. The attachment of the eight cable assemblies to the outriggers of the 1/5-scale model is shown in figure 12, and a view of a typical springbank and leveling device (turnbuckles were used on the model) is shown in figure 16. Two factors were considered in designing this suspension to simulate the suspension used to support the full-scale vehicle. First, it was considered important to support the model at the ends of the outriggers as the full-scale vehicle was supported, so that the load paths would be the same for the model and full-scale vehicles. Second, it was considered important to maintain on the model the same ratio of first-bending-mode frequency to the rigid-body rocking-mode frequency as existed on the full-scale vibration tests. Model vibration tests were made with the two values of stiffness of the eight-cable system, one value obtained with the springs in the spring bank and the other value obtained by replacing the springs by rigid steel links. In all tests the model was supported by cables attached to each of the eight outriggers. The vibration investigation of the model with this suspension system is described in reference 2.

COMPARISONS OF VIBRATION CHARACTERISTICS OF MODEL AND FULL-SCALE VEHICLES

Principal Bending-Mode Frequencies

The first four experimentally determined principal resonant frequencies of the full-scale Saturn vibration test vehicle are compared with the frequencies of the 1/5-scale Saturn model in figure 17. The comparison is made on the basis of the frequency parameter \bar{f} which is obtained, in accordance with the scaling relation, by multiplying the frequency in cycles per second by the respective vehicle length. In the ideal situation where all significant effects have been properly scaled, the model and full-scale frequency parameters would be identical. The frequencies of the full-scale vehicle used for figure 17 were obtained with the soft suspension. The first-bending-mode frequency of the model was obtained with the eight-cable link suspension, but the higher model frequencies are from either the spring or link suspensions, because, as shown in reference 2,

this suspension change caused only small changes of the higher mode frequencies. Figure 17 shows good agreement between the model and full-scale first-bending-mode frequency parameters; this agreement indicates that all significant parameters affecting the first-bending-mode frequency have been properly scaled. The first and second cluster mode frequencies of the model are about 10 percent below those of the full-scale vehicle, and greater differences between the model and full-scale second-bending modes are indicated. These differences between the model and full-scale higher modes are thought to be primarily due to the omission of ring frames from the second stage of the Saturn model, which resulted in coupling between the shell mode of the second-stage outer shell and the bending modes.

The variation of the first-bending-mode frequency parameter of the model and full-scale Saturn vehicles with booster water level is shown in figure 18. It should be noted that the frequency parameter scale has been greatly expanded for this figure so that the secondary effects can be more easily seen. First-bending-mode frequency parameters are presented in figure 18(a) for booster water levels from zero to 100 percent full, representing the burnout to lift-off flight conditions; and for three suspension-system configurations for the model and for two suspension-system configurations for the full-scale vehicle. As an indication of the stiffness of the various suspension-system configurations, the rigid-body rocking-frequency parameter is presented in figure 18(b).

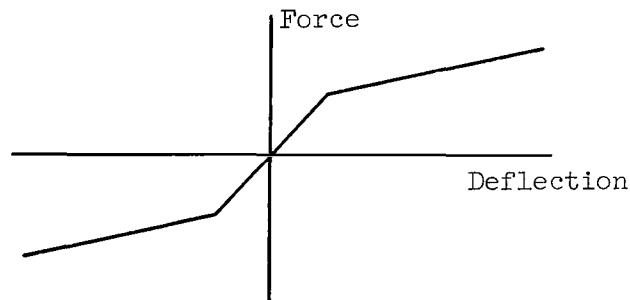
Figure 18(b) shows that the rigid-body rocking-frequency parameter (which increases with increasing stiffness of the suspension) of the 1/5-scale model was smallest with the two-cable suspension (diamond symbols), larger with the eight-cable-spring suspension (triangular symbols), and largest with the eight-cable-link suspension (circular symbols) where its value falls between the values obtained with the two suspension configurations of the full-scale vehicle. Figure 18(a) shows that the first-bending-mode frequency parameter of the model for small force values was smallest with the two-cable suspension, larger with the eight-cable-spring suspension, and largest with the eight-cable-link suspension where for booster water levels of less than 100 percent full its value is within about 3 percent of the value obtained from the full-scale vehicle with the soft suspension. At the 100-percent-full booster water level, the model frequency parameter is about 6 percent above the full-scale value. This disagreement is thought to be due to strong coupling of the first bending mode with the outer tank modes. Evidence of this coupling is shown by the multi-peaked frequency-response curves at this water level shown in references 2 and 3, and by the relatively large difference between the center tank deflection and the outer tank deflection shown in figure 20 for the 100-percent-full water level. Thus, figure 18 shows that the first-bending-mode frequency parameters of the model was in good agreement with the full-scale Saturn vehicle values when the model suspension stiffness has been properly scaled. The suspension system was considered to be properly scaled when the rocking frequency parameters of the model and full-scale vehicles were in agreement.

An effect of shaker force magnitude on the first-bending-mode frequency parameter of the 1/5-scale model with the eight-cable suspension is shown in figure 18(a). At the 75- and 100-percent-full booster water levels values of the frequency parameter are shown for shaker input forces of less than 18 pounds (the open circular or triangular symbols), and for 38 pounds (the solid circular

or triangular symbols). Comparison of the frequencies obtained with 38 pounds force (the solid symbols) with those obtained with less than 18 pounds force (the open symbols) shows that the effect of increasing the shaker input force was to decrease the first-bending-mode frequency parameter; this decrease indicates that the vibratory response of the Saturn model is nonlinear.

Vibratory Nonlinearity of Saturn

The linearity of the response of the Saturn vehicle, both model and full scale, has been investigated and the results are presented in figure 19. (These data were obtained from references 1, 2, and from unpublished data furnished by the Marshall Space Flight Center.) In figure 19 the variation of \bar{x} (the peak nose deflection divided by vehicle length) with the frequency parameter \bar{f} is presented. Each data point on this figure was obtained by varying the frequency of the shaker input force while holding its magnitude constant until maximum response of an accelerometer at the nose of the vehicle in the first bending mode was obtained; thus each curve shown in this figure represents the locus of the peaks of a family of curves of deflection as a function of frequency where each such curve is obtained for a different shaker force magnitude. The variation of \bar{x} with \bar{f} of the 1/5-scale Saturn model with five combinations of weight condition and suspension configuration is shown in figure 19(a). The curves in figure 19(a) show that the frequency \bar{f} decreases as the deflection \bar{x} increases. In systems having linear springs the curves showing the variation of \bar{x} with \bar{f} (such as shown in fig. 19) would be straight vertical lines; therefore, the curves in figure 19 showing the greatest variation of \bar{f} with \bar{x} indicate the system with the greatest nonlinearity. This variation is the type associated with a "softening" spring, which has a force-deflection curve such as shown in the following sketch:



The nonlinearity occurs at all three weights and for all three suspensions and indicates that the nonlinear behavior is not solely a property of the suspension or of a particular weight condition. The variation of \bar{x} with \bar{f} for the full-scale Saturn Block II vehicle is shown in figure 19(b). (The Saturn Block II vehicle structure is described in reference 9 and is different from the Block I structure in several areas; however, the clustered tank configuration and the riveted type of construction are the same for both Block I and Block II vehicles.) The variation of \bar{x} with \bar{f} for the full-scale Saturn shown in figure 19(b) is the same, although less pronounced, as the variation shown in figure 19(a) for the 1/5-scale model; thus, the nonlinear behavior is not solely a property of the 1/5-scale model.

The effect of the preload in the outer lox tank upper joints has been investigated as a possible source of nonlinearity and the results, obtained from reference 1, are presented in figure 19(c). The variation of \bar{x} with \bar{f} is presented for the 1/5-scale Saturn model with the two-cable suspension and with 0 and 900 pounds preload in the outer lox tanks. Figure 19(c) shows that the nonlinearity is present for both values of preload, and that the effect of increasing the preload from 0 to 900 pounds was to decrease, but not eliminate, the nonlinearity. Thus, figure 19(c) shows that the preload in the outer lox tanks has a large effect on the nonlinear behavior of the Saturn model. Two effects of increasing the outer tank preload are to decrease the amount of "slop" in the riveted joints, and to increase the bending required to "unseat" the outer lox tank upper joints (which cannot transmit tension forces). Either of these effects is a possible cause of the nonlinearity. Other possible sources of nonlinearity are liquid sloshing, nonlinear damping, nonlinear stiffness associated with structure such as the second-stage adapter or the third-stage conical shell adapter, and nonlinear transmission of shaker force through such structure as the outer-tank attachment joints.

Mode Shapes

Model and full-scale mode shapes are compared in figures 20 to 23. The first bending modes are shown in figure 20 to be in good agreement for the three weight conditions. Following the convention of reference 1, a mode where the outer-tank deflections are generally of about the same magnitude and phase as the center tank is termed a bending mode. There is a difference, however, between the deflections of the outer fuel tank of the model and full-scale vehicles at the 100-percent-full weight condition. This difference, together with the fact that the model outer fuel tank mode occurred at a lower frequency than the model first bending mode whereas the full-scale outer-fuel-tank mode occurred at a higher frequency than the first bending mode, indicates that some structural differences between the model and full-scale vibration vehicle configurations exist. It is possible, for example, that because the outer-fuel-tank connecting joints offer very little rotational restraint to the tank ends, the added stiffness of the aerodynamic fairing and fuel piping present on the full-scale vehicle might be sufficient to cause these mode-shape differences.

Model and full-scale first and second cluster modes are shown in figures 21 and 22, respectively. Again following the convention of reference 1, modes where the outer tank deflection are generally in the opposite direction as those of the center tank are termed cluster modes. In all the modes shown herein, there is good qualitative agreement between the model and the full-scale mode shapes; that is, the sign of the deflections of the outer tanks relative to the center tank is the same for both model and full-scale vehicles. In addition, the model and full-scale center-line deflections, indicated by the circular symbols, are generally in good agreement. Figures 21 and 22 show that the differences between the deflections of the model and full-scale outer tanks are larger for these modes than for the first bending mode, and that large differences of slope exist between the outer tanks and the center line at the points where they join. These differences suggest again that the aerodynamic

fairings and fuel piping which were not included on the model might have some effect on the mode shapes.

The model and full-scale second bending modes are compared in figure 23 for two weight conditions. The disagreement between the model and full-scale mode shapes shown in this figure is thought to be caused by the omission of the ring frames in the second stage of the model. The deflection of the center tank of the second stage of the model, indicated by the triangular symbols, is shown to be considerably different from the deflection of the outer shell, indicated by the flagged circular symbols between stations 1000 and 1400. The mode shape of a cross section through the second stage of the model at station 210 (equivalent to the full-scale station 1050) is shown in figure 24. Figure 24 shows that the outer shell is vibrating in a shell mode with seven waves around the circumference while the center tank is translating with an essentially undeformed cross section. The deflections at points A and B, which are the circumferential locations where the mode shapes shown in figure 23 were measured, are shown to be in opposite directions. There are no indications that such a shell motion existed on the full-scale vehicle.

Damping

Model and full-scale damping values g are compared in table I. Damping values obtained from the full-scale vibration vehicle SA-D1, the 1/5-scale model with the two-cable suspension, and the 1/5-scale model with the eight-cable suspension are given for four weight conditions and the first four modes. It can be seen that two values of damping are given in some cases for the model. These values of damping were obtained for different values of the vibration amplitude; the value on the left is for large amplitude and is usually larger than the value on the right which was obtained for smaller amplitudes. This figure shows that the model and full-scale damping values for the first bending mode are in good agreement, except for the value of 0.116 given for the full-scale vehicle at 75 percent full. This high value of damping is explained in reference 3 on the basis of outer-tank coupling. For the higher modes, the agreement is not as good as for the first bending mode; however, the model and full-scale damping are usually of the same order of magnitude.

CONCLUDING REMARKS

An investigation of the vibration characteristics of a 1/5-scale dynamic replica model of Saturn SA-1 has been performed at Langley Research Center with two suspension systems. Resonant frequencies, mode shapes, and damping from this model are compared herein with the results of a ground vibration survey of a full-scale Saturn vehicle conducted at George C. Marshall Space Flight Center. Comparisons of the model results with the full-scale results show the following:

(1) The model and full-scale first-bending-mode frequency parameters are in good agreement (within 6 percent) when the rigid-body suspension-system rocking frequency parameters are in agreement. The frequency parameters of the

model in the first cluster mode, the second cluster mode, and the second bending mode are approximately 10 percent below the corresponding full-scale frequency parameters.

(2) For most of the modes the damping of the model is the same order of magnitude as the damping of the full-scale Saturn.

(3) The mode shapes of the model in the first bending, first cluster, and second cluster modes are in agreement with the corresponding full-scale mode shapes. Significant differences between the model and full-scale second-bending-mode shapes are believed to be caused by a structural simplification made in the second-stage structure of the model.

(4) Both model and full-scale vehicles exhibit a nonlinearity of the first-bending-mode response characterized by a decrease of resonant frequency with increase of vibration amplitude. For the range of amplitudes investigated with the model, the variation of frequency was approximately the same as the variation caused by suspension-system stiffness changes.

Langley Research Center,
National Aeronautics and Space Administration,
Langley Station, Hampton, Va., June 4, 1964.

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TABLE I.- COMPARISON OF DAMPING FACTORS OBTAINED FROM FULL-SCALE AND 1/5-SCALE MODEL OF SATURN SA-1

$$\left[g = \frac{1}{2\pi n} \log_e \frac{x_n}{x_0} \right]$$

Configuration	Damping factors for -						
	Booster tank empty (burnout)		48 percent full (maximum dynamic pressure)		75 percent full (35 sec)	100 percent full (lift-off)	
	Soft suspension	Stiff suspension	Soft suspension	Stiff suspension	Soft suspension	Soft suspension	Stiff suspension
First bending mode							
Full-scale Saturn	0.024	----	0.023		0.116	0.026	-----
1/5-scale model - 2 cable	.030, .017		.032, .017		.020	.033, .025	
1/5-scale model - 8 cable	.032, .015	0.03	.030, .013	0.033, .012	.037, .015	.030	0.01
First cluster mode							
Full-scale Saturn	-----	----	0.024		0.028	0.026	-----
1/5-scale model - 2 cable	-----		.023, .011		.025, .015	.017	
1/5-scale model - 8 cable	-----	----	-----	0.030, .013	-----	-----	-----
Second cluster mode							
Full-scale Saturn	-----	----	-----	-----	0.021	0.022	-----
1/5-scale model - 2 cable	-----		-----		.022, .016	.014	
1/5-scale model - 8 cable	-----	----	-----	-----	.012	.015	0.018
Second bending mode							
Full-scale Saturn	0.018	----	-----	-----	0.010	0.010	-----
1/5-scale model - 2 cable	.046, .032	----	-----		.039, .019	.010	
1/5-scale model - 8 cable	-----	----	-----	-----	.017	-----	-----

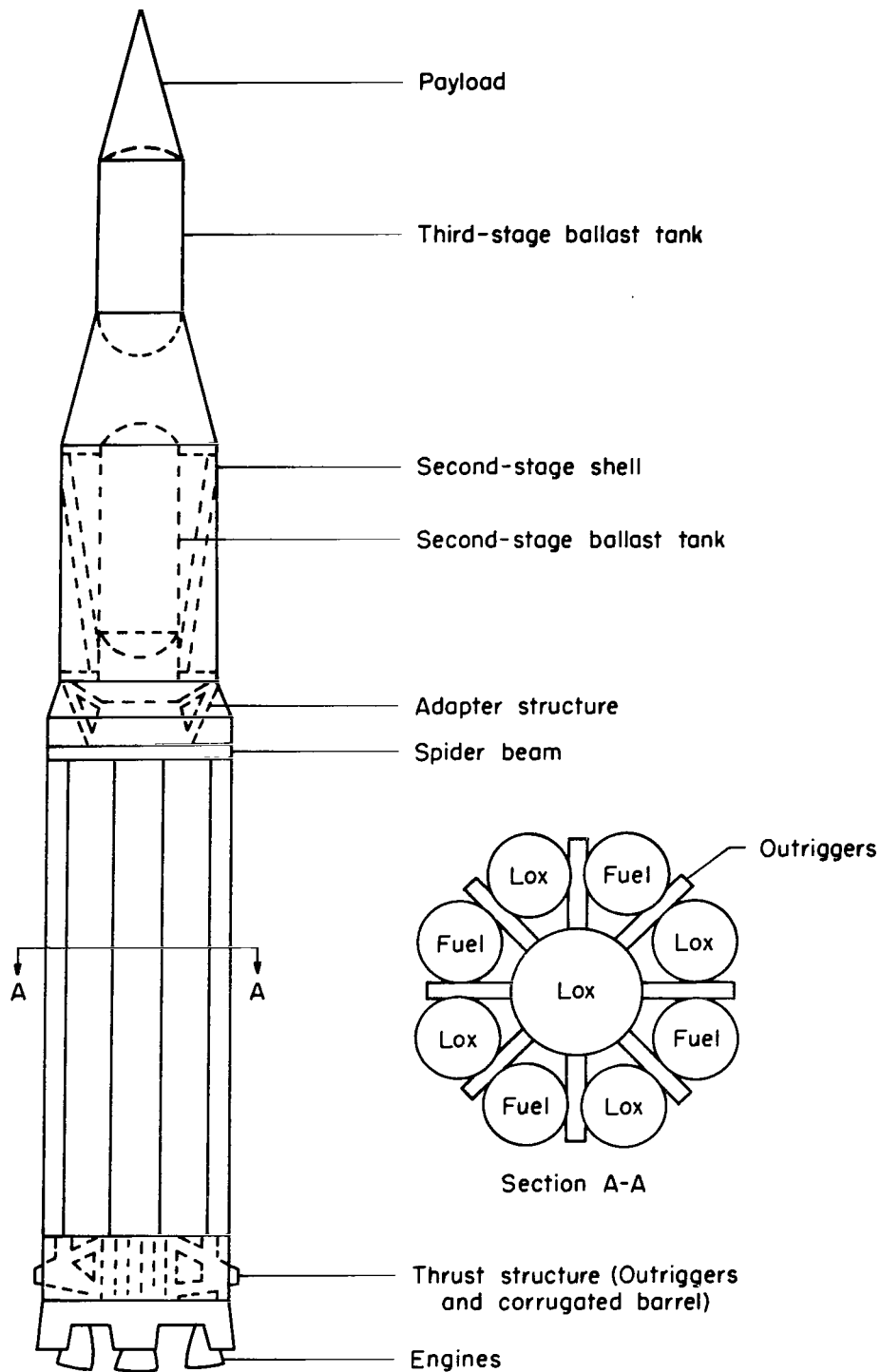


Figure 1.- Sketch of Saturn block I vehicle.

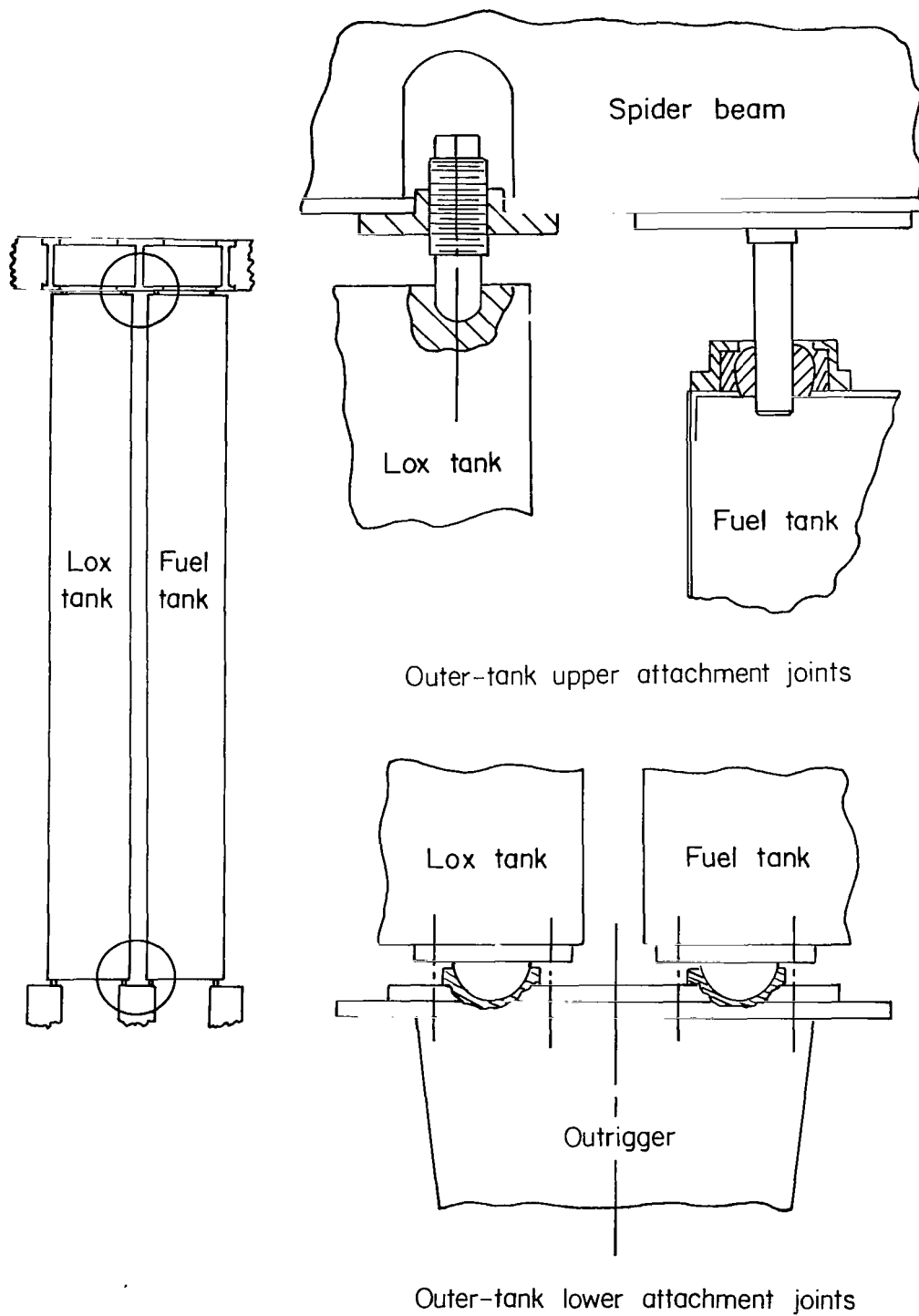
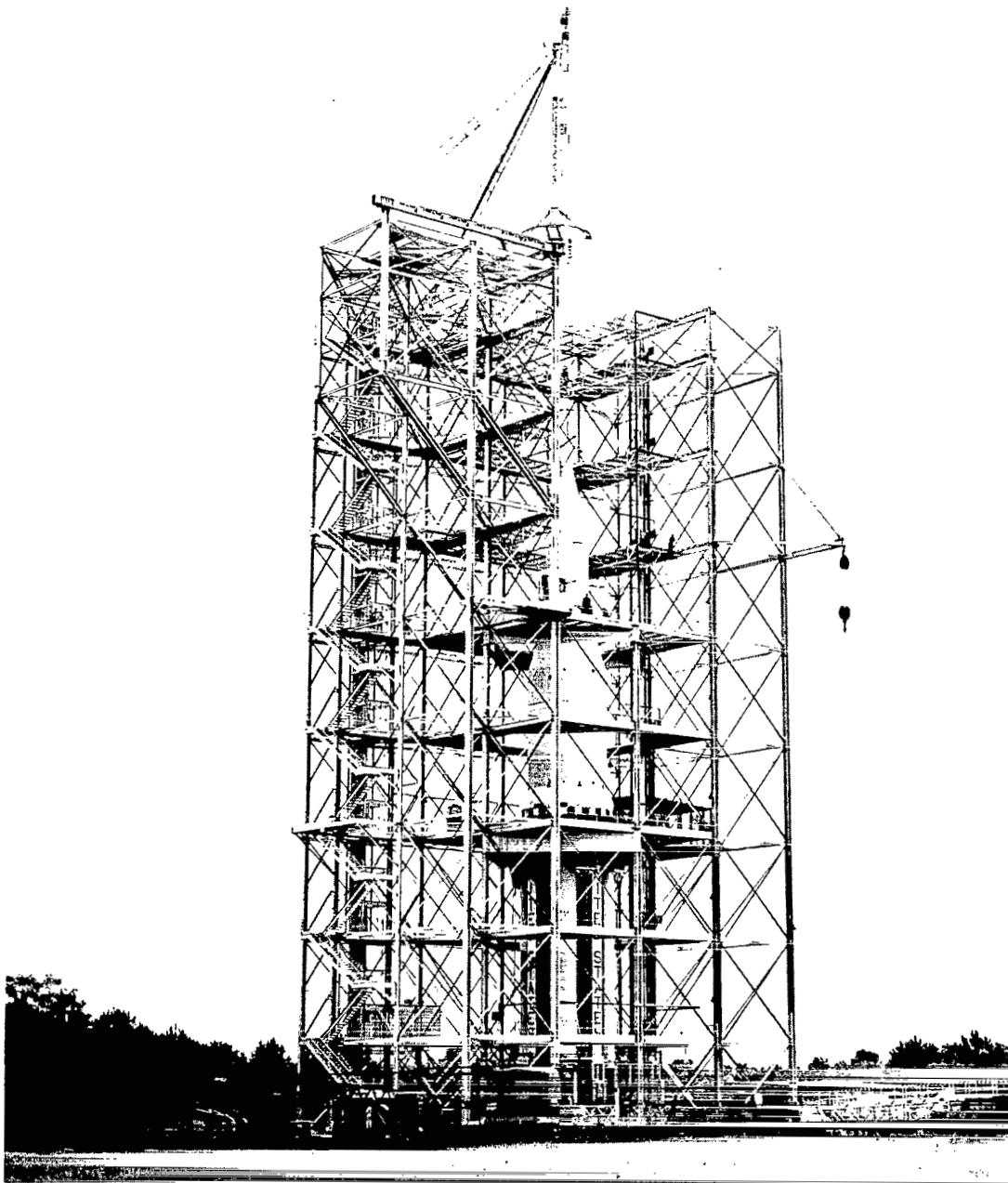


Figure 2.- Sketch of outer-tank attachment joints.



M-61-1292-6

Figure 3.- Full-scale Saturn vibration test vehicle at Marshall Space Flight Center.

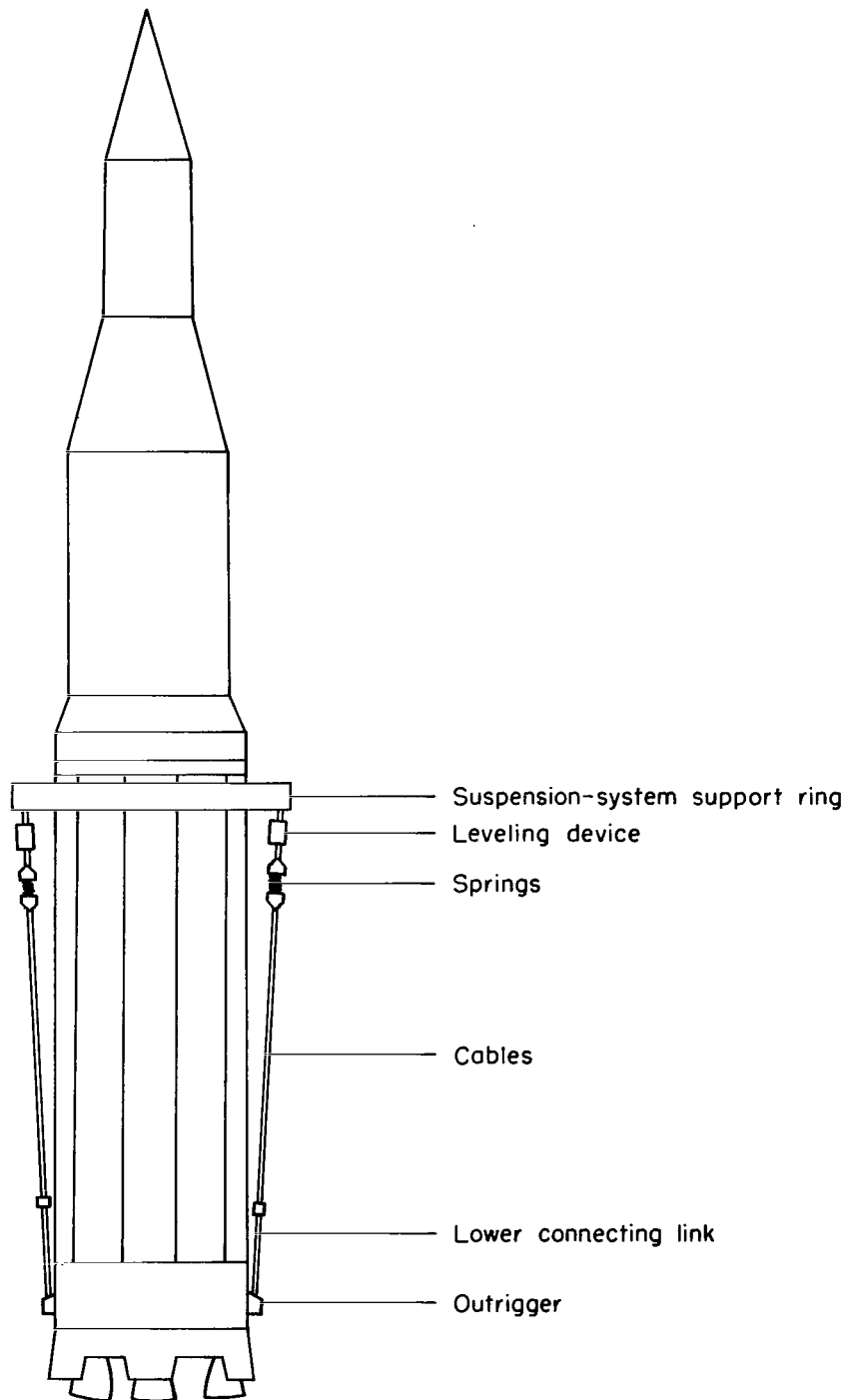


Figure 4.- Sketch of Saturn eight-cable suspension.

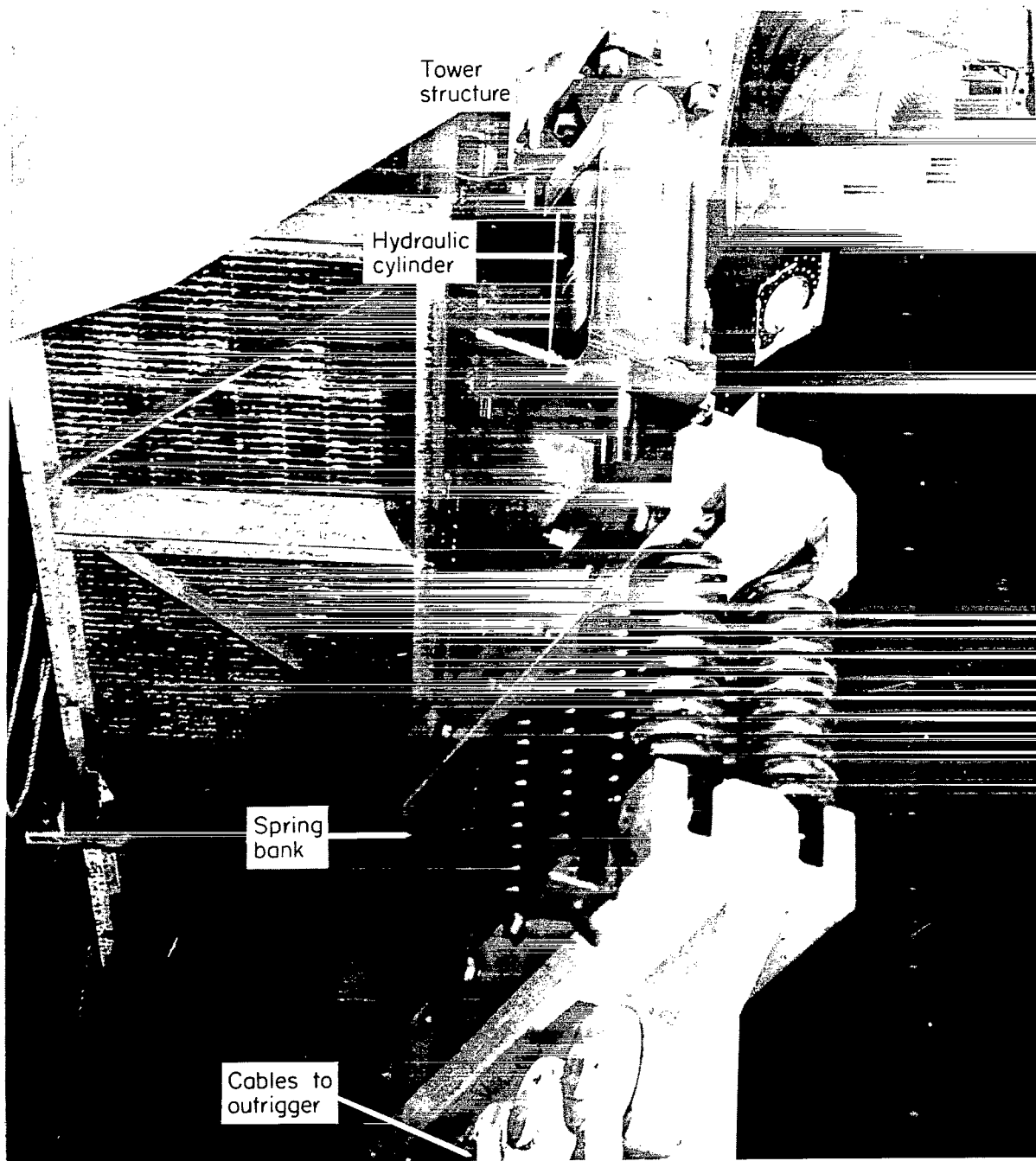


Figure 5.- Spring bank area of full-scale Saturn.

L-64-4797

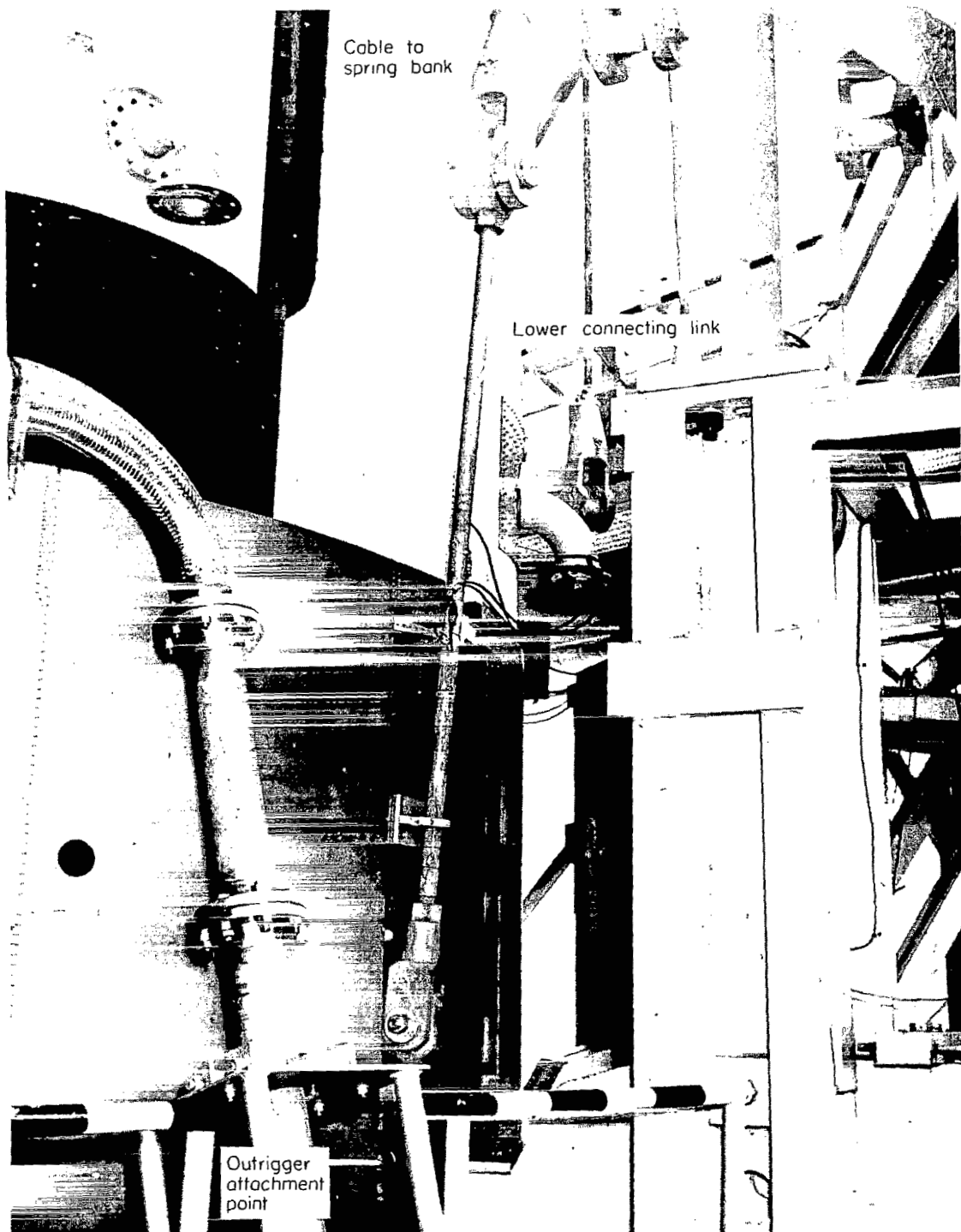


Figure 6.- Lower connecting links of full-scale Saturn.

L-64-4798

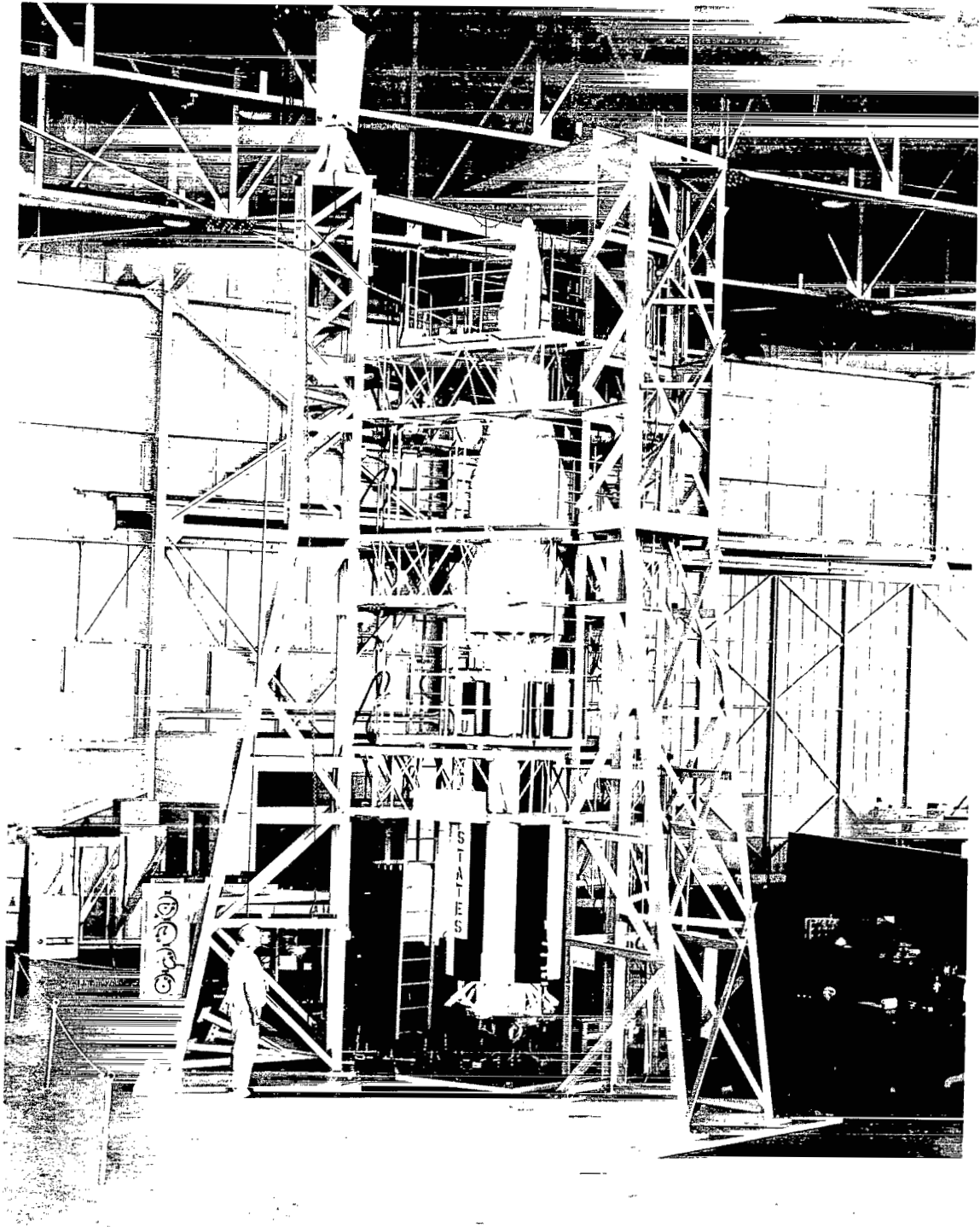


Figure 7.- 1/5-scale model of Saturn SA-1 in vibration testing tower at Langley Research Center. L-61-4079

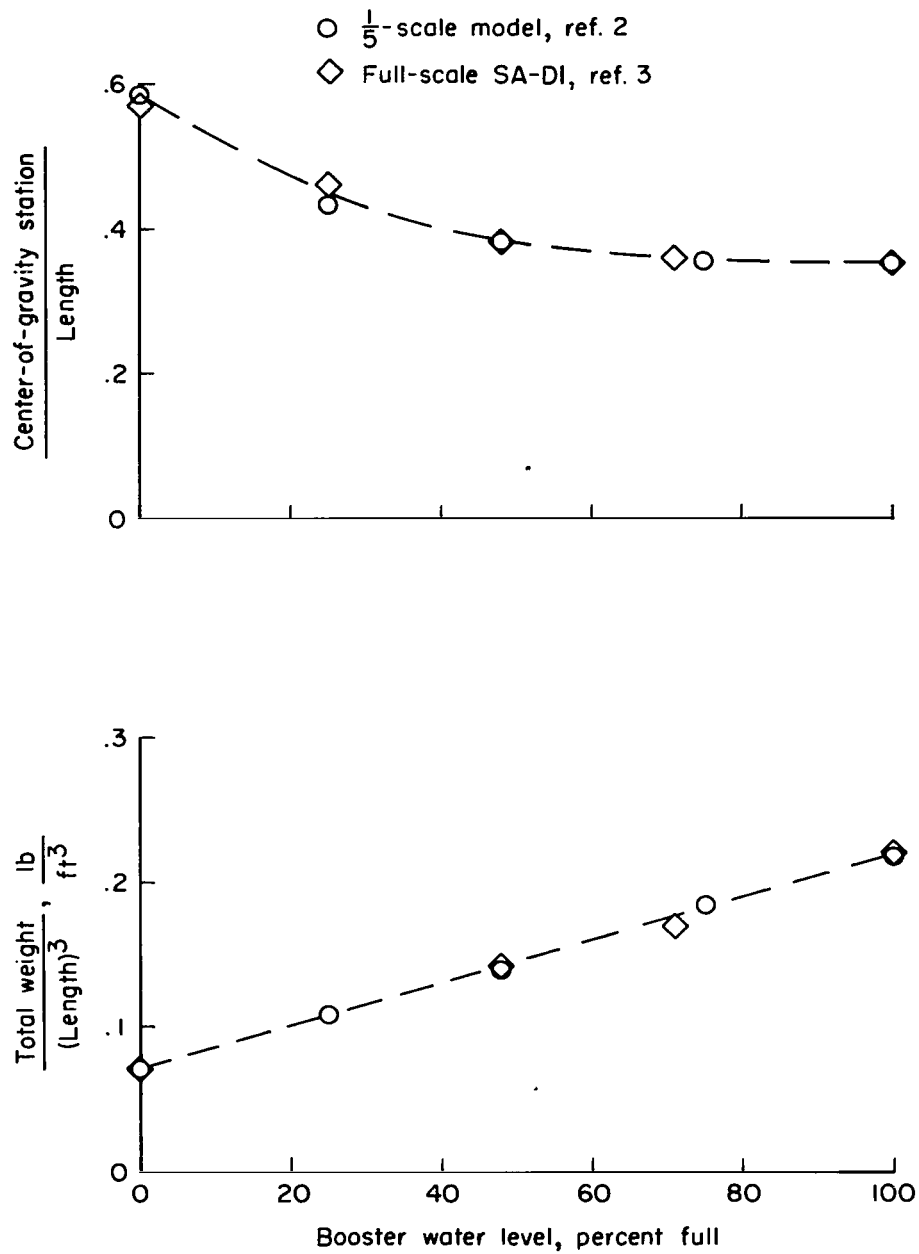


Figure 8.- Comparison of model and full-scale Saturn weight and center-of-gravity location.

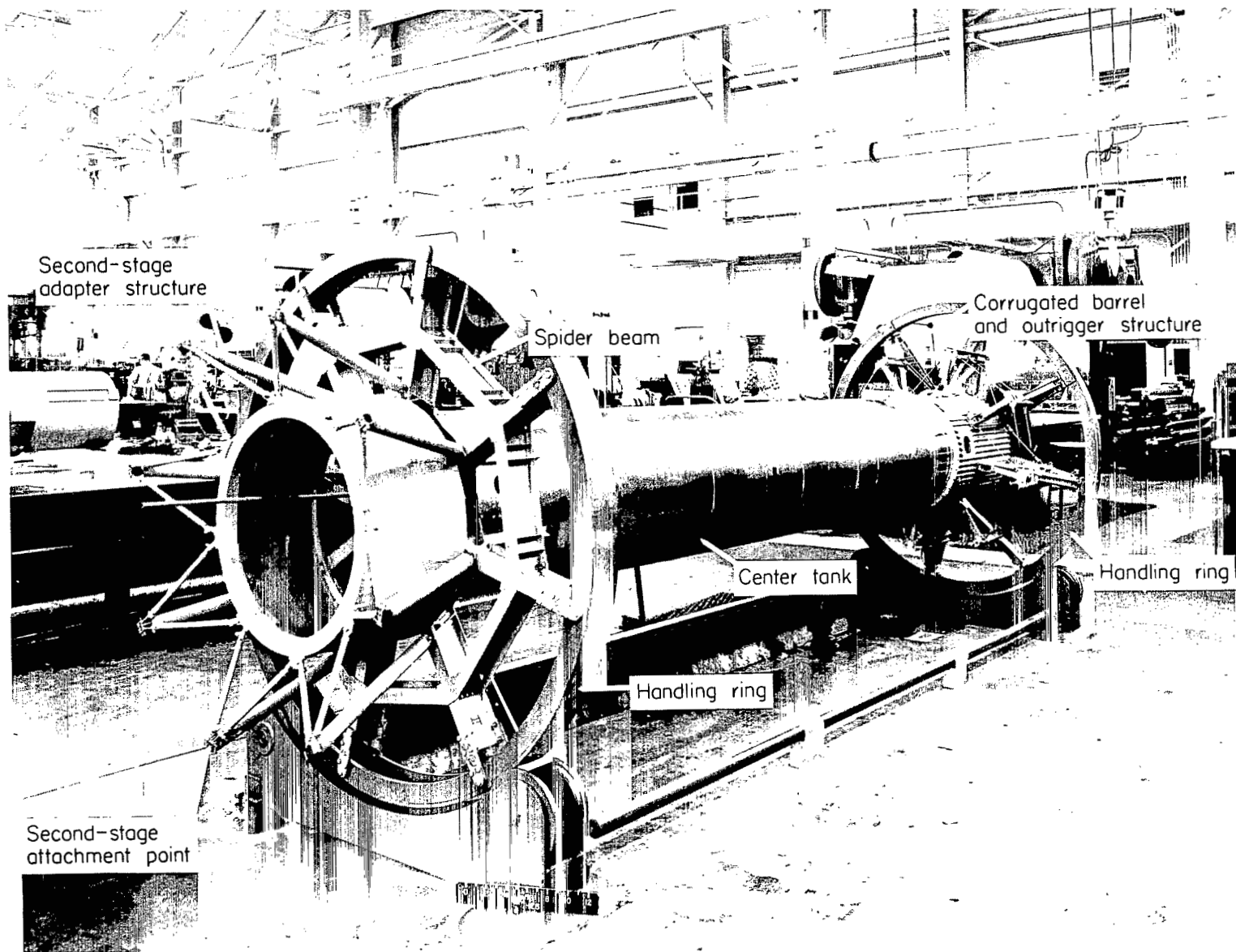


Figure 9.- Model booster structure during assembly.

L-61-1654.1

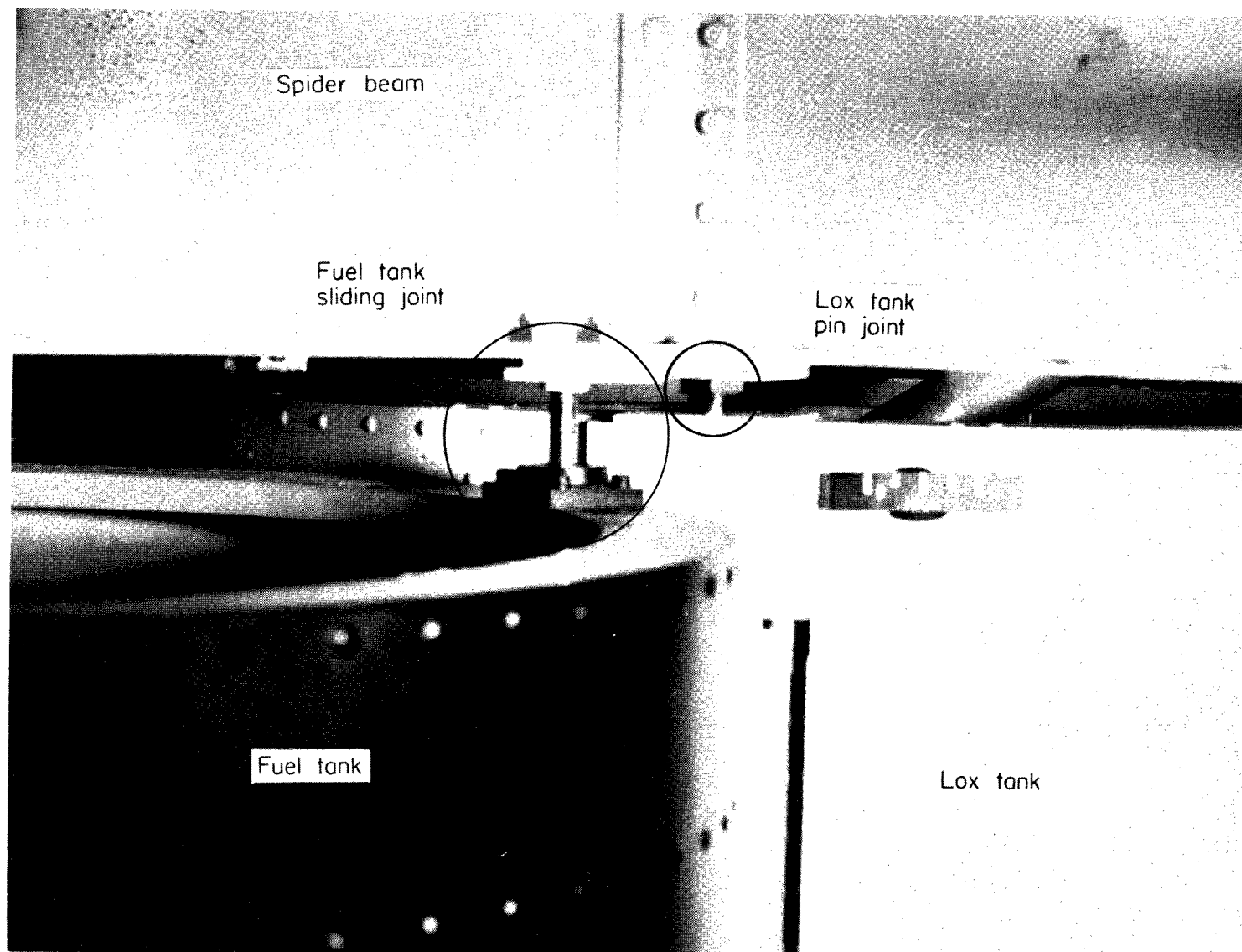
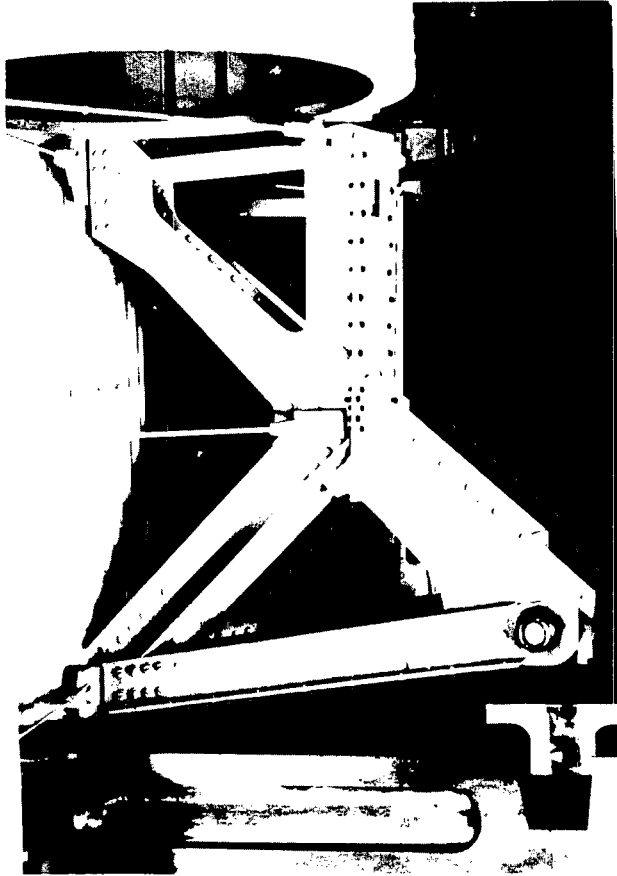
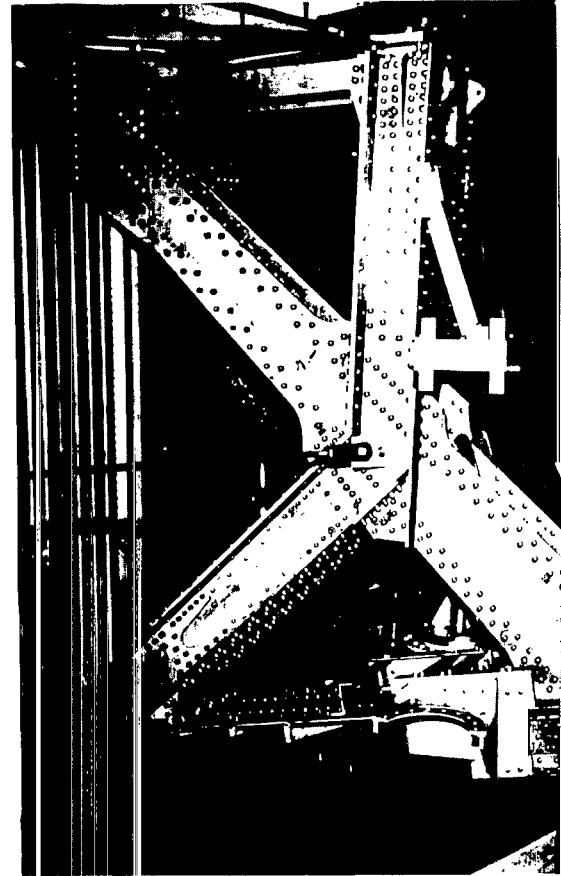


Figure 10.- Upper attachment joints of outer tanks of 1/5-scale Saturn model. L-62-3336.2



Model



Full scale

Figure 11.- Model and full-scale Saturn outrigger structure.

L-62-1011.1

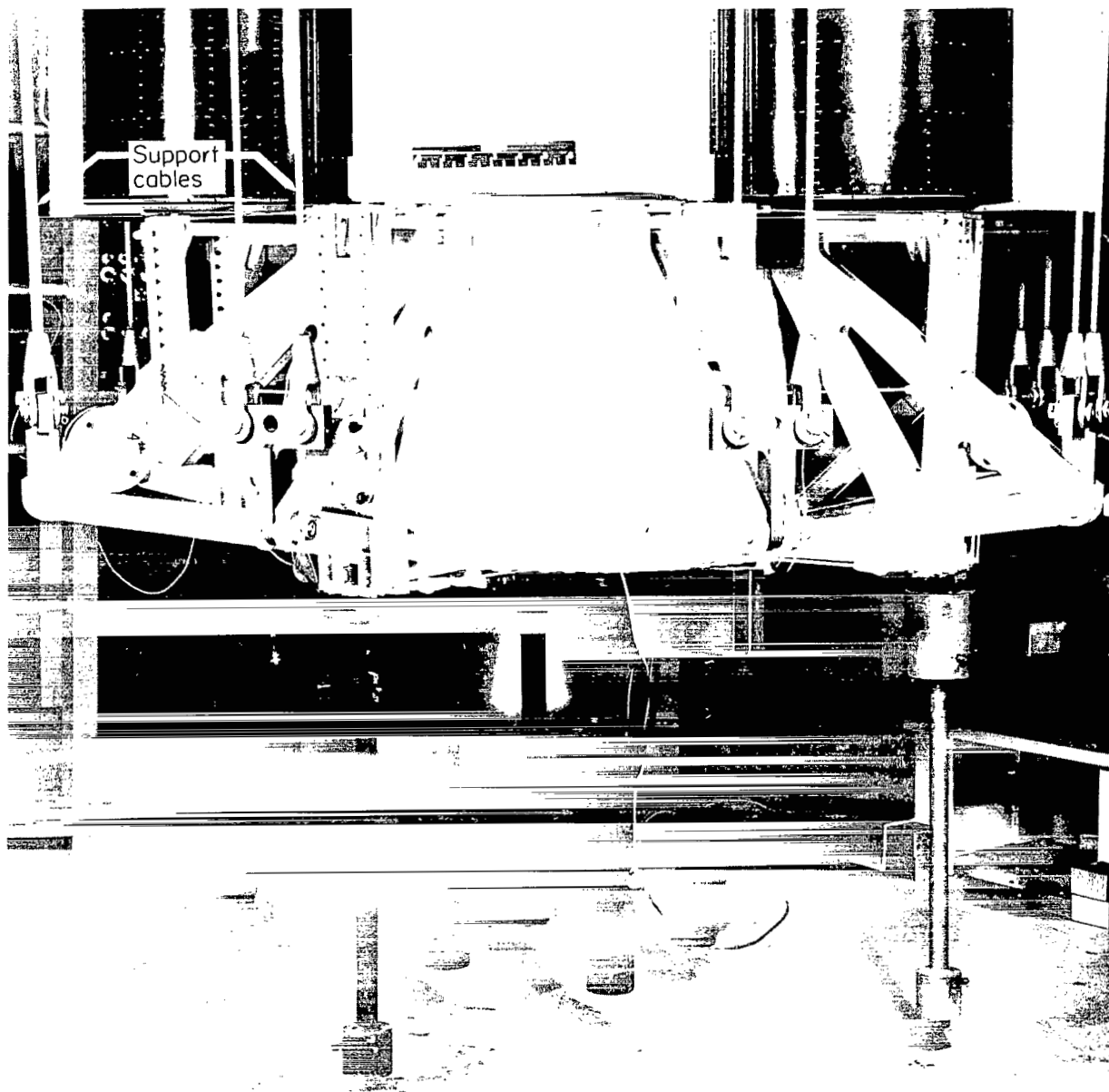


Figure 12.- Thrust structure and engines of 1/5-scale Saturn model with eight-cable suspension system. L-62-5095.2

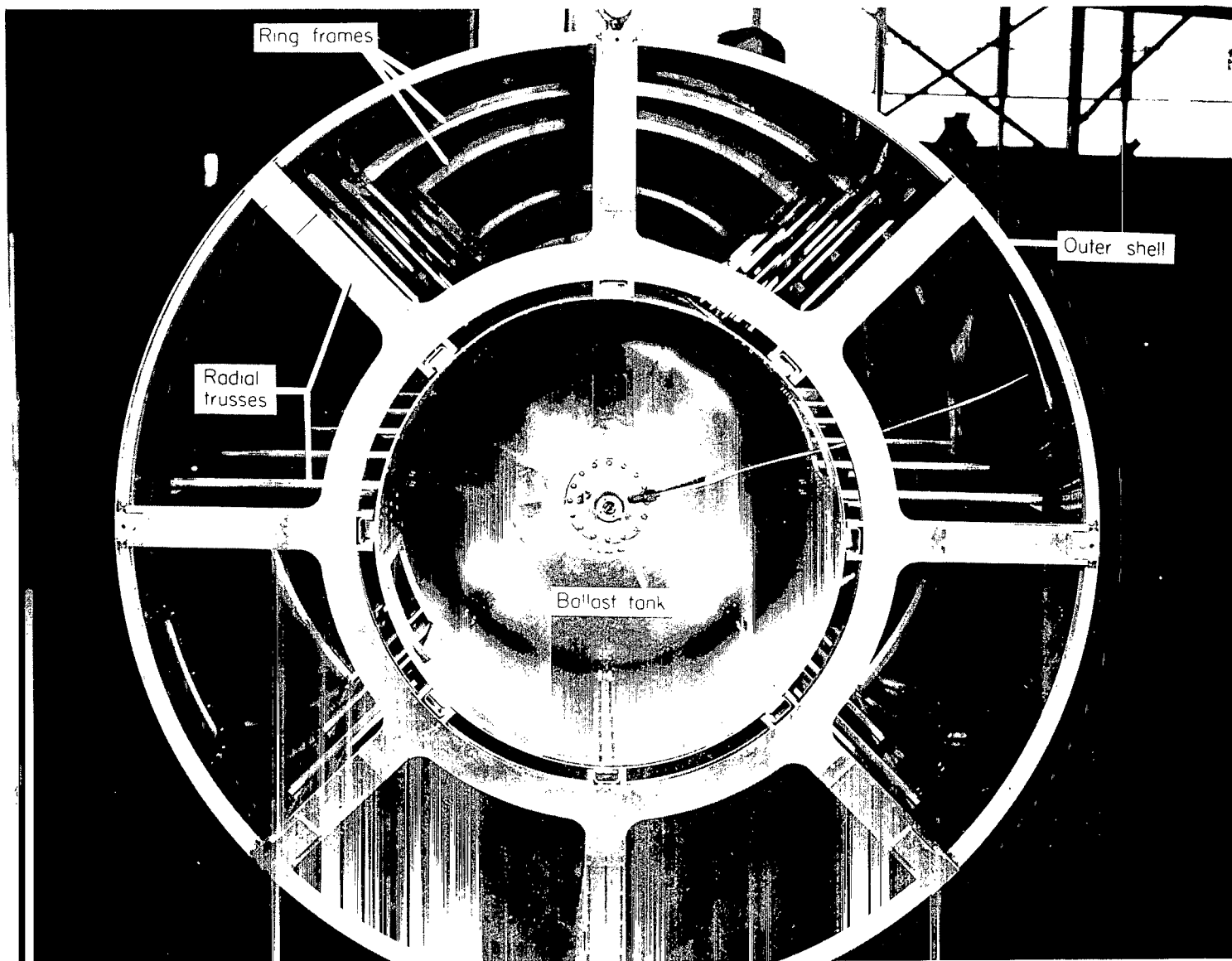


Figure 13.- End view of second stage of 1/5-scale Saturn.

L-61-2988.2

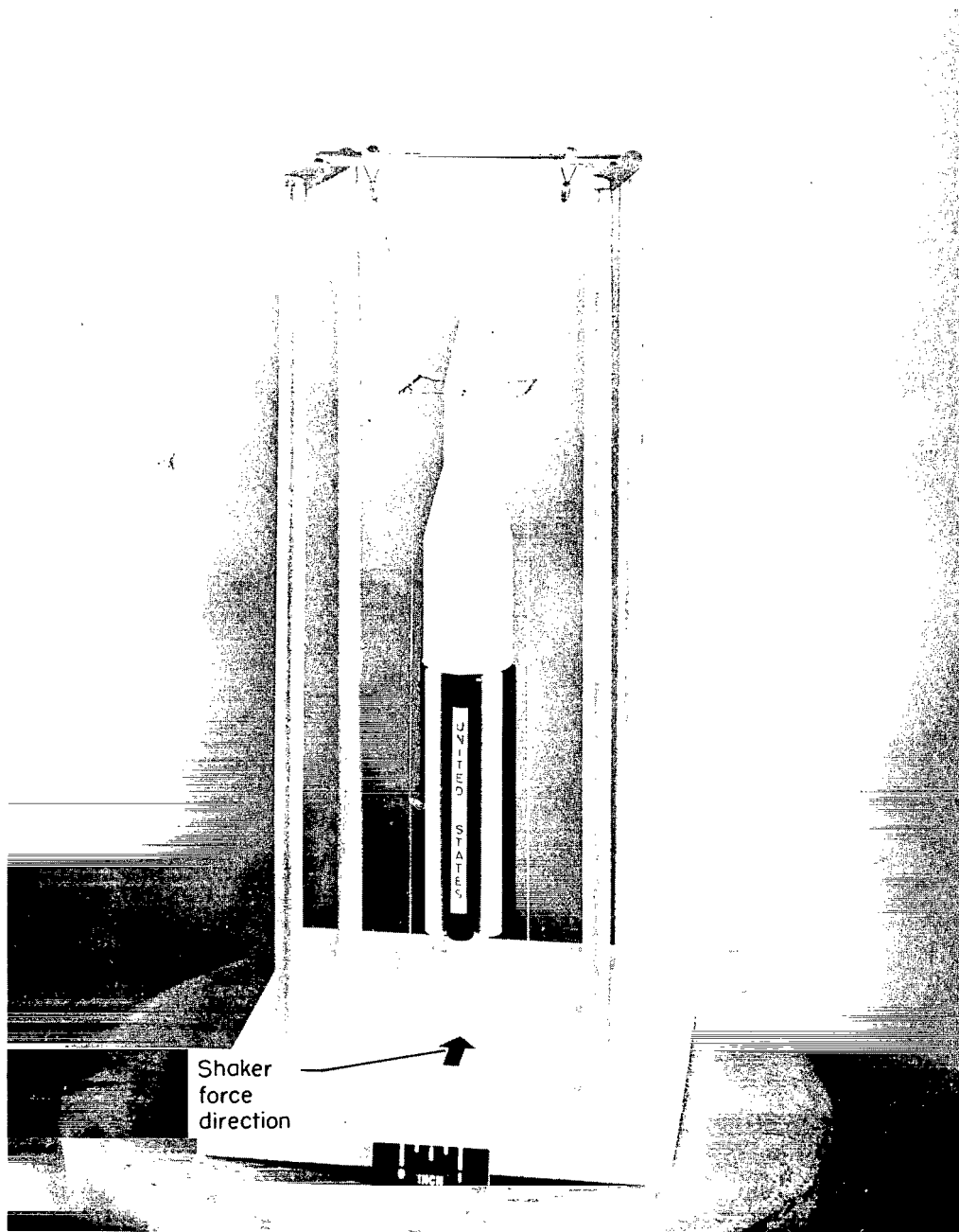


Figure 14.- Two-cable suspension-system display model. L-61-6581.1

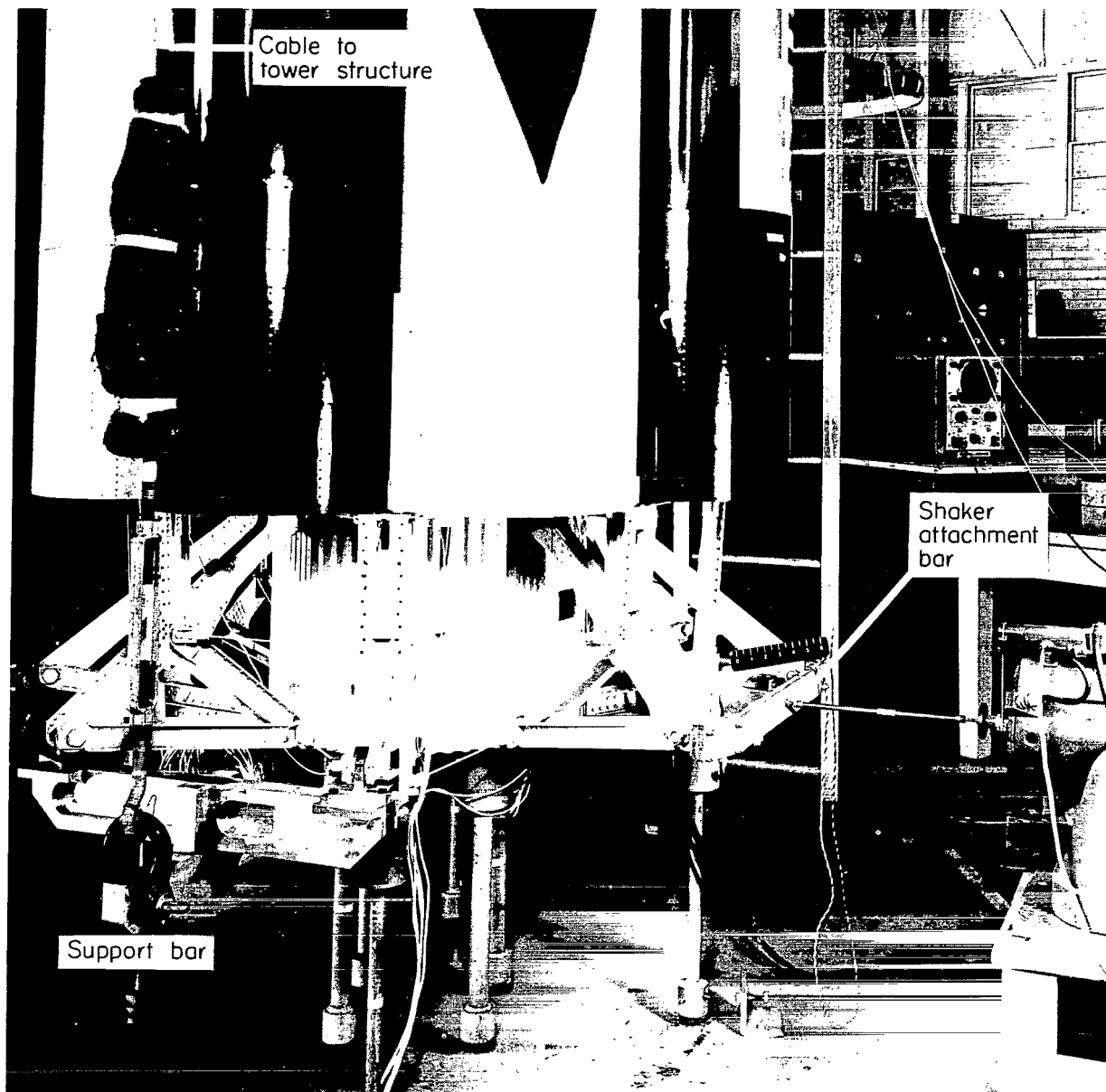


Figure 15.- Base of Saturn model with two-cable suspension system.

L-64-4799

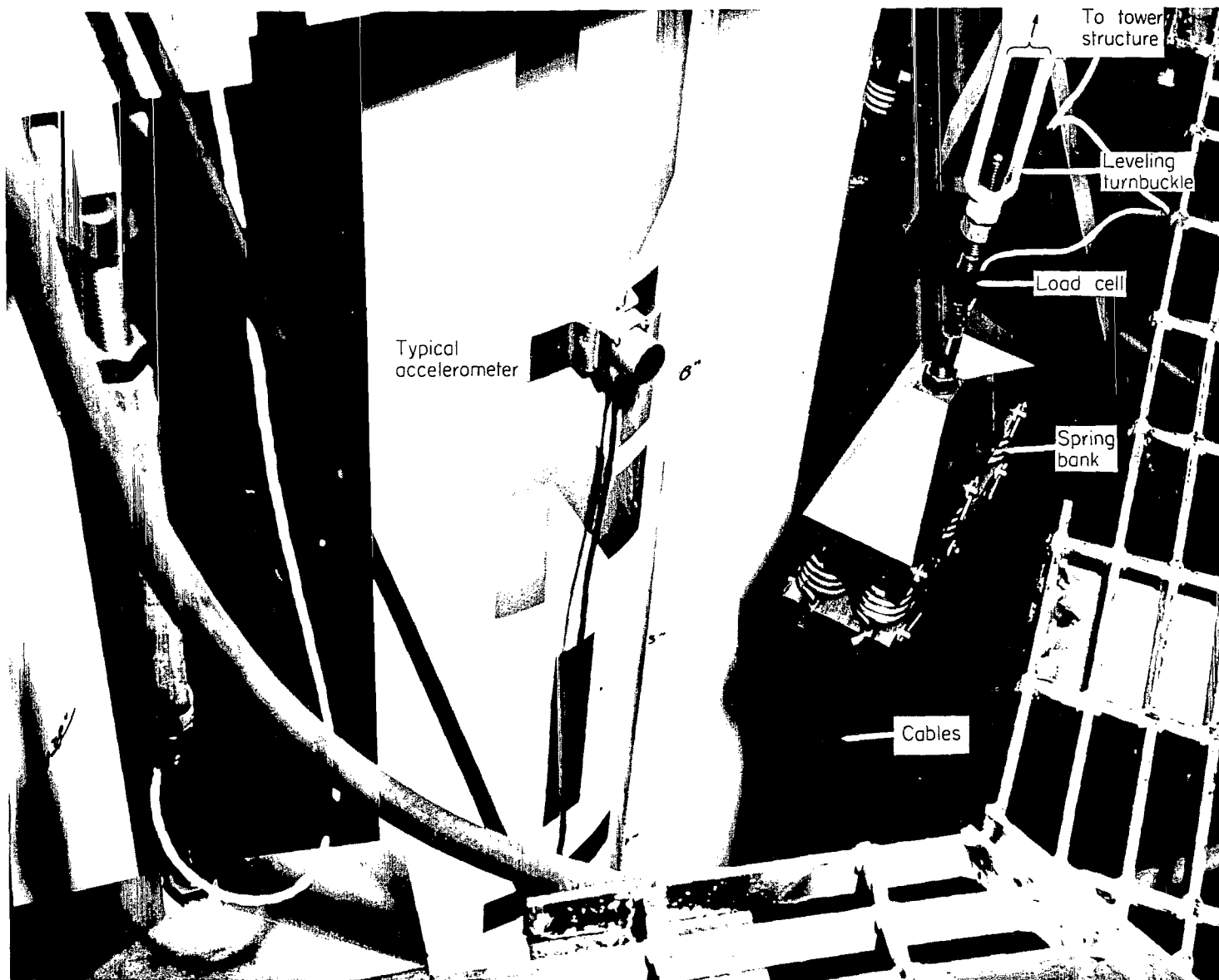


Figure 16.- Spring bank area of 1/5-scale Saturn model.

L-62-5099.2

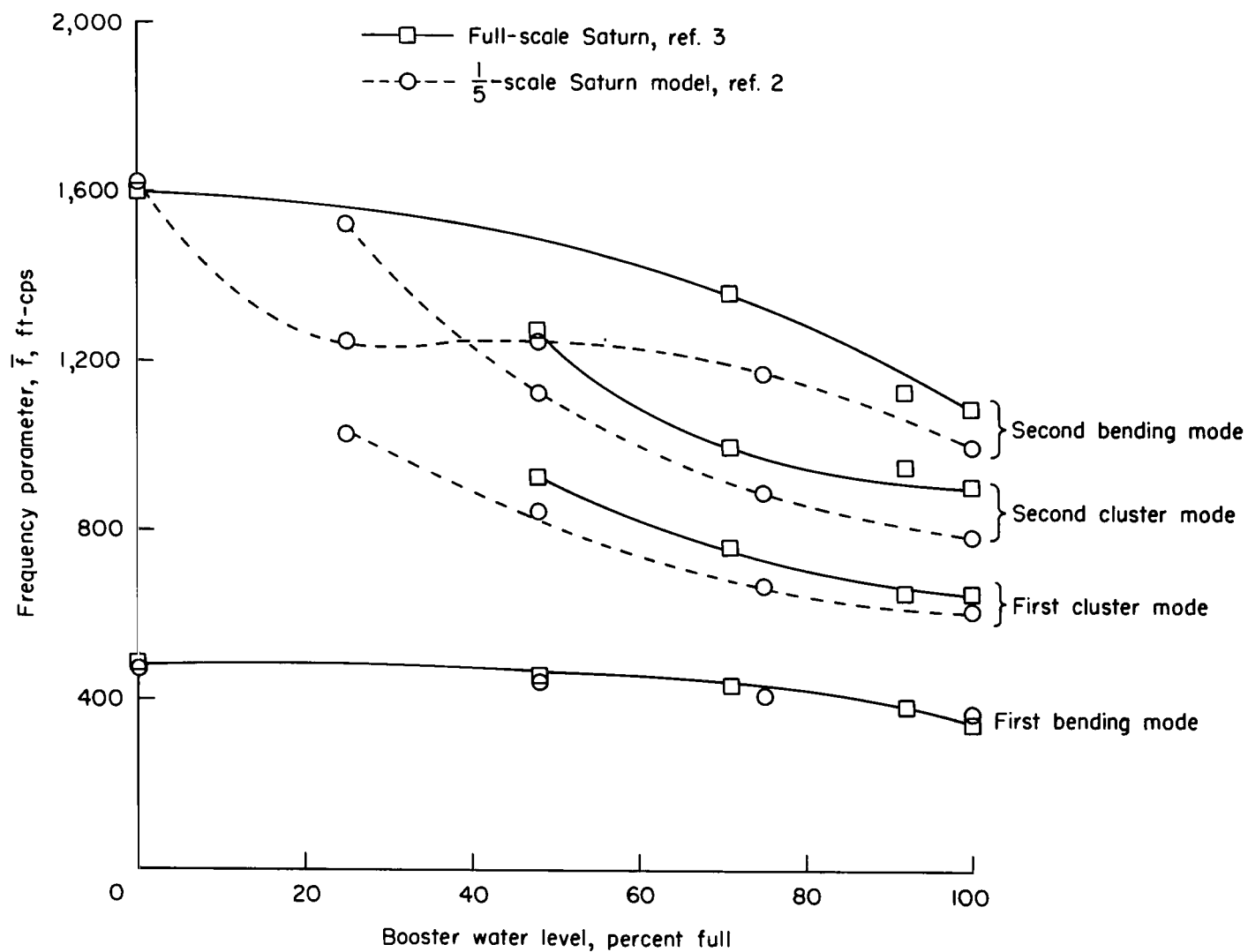
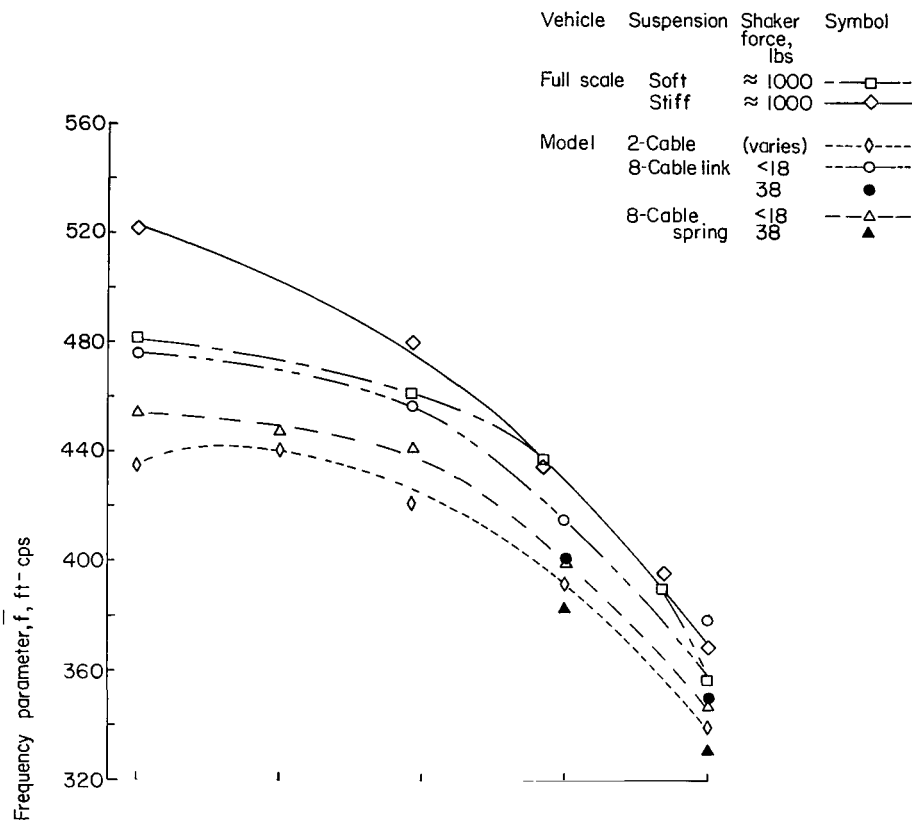
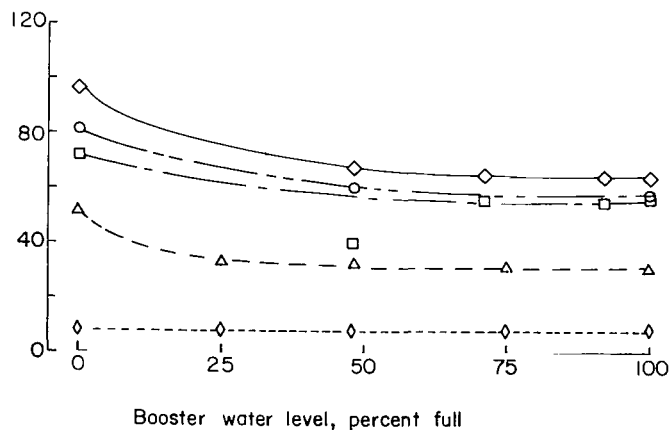


Figure 17.- Variation of frequency parameter of first four principal modes of Saturn with booster water level.

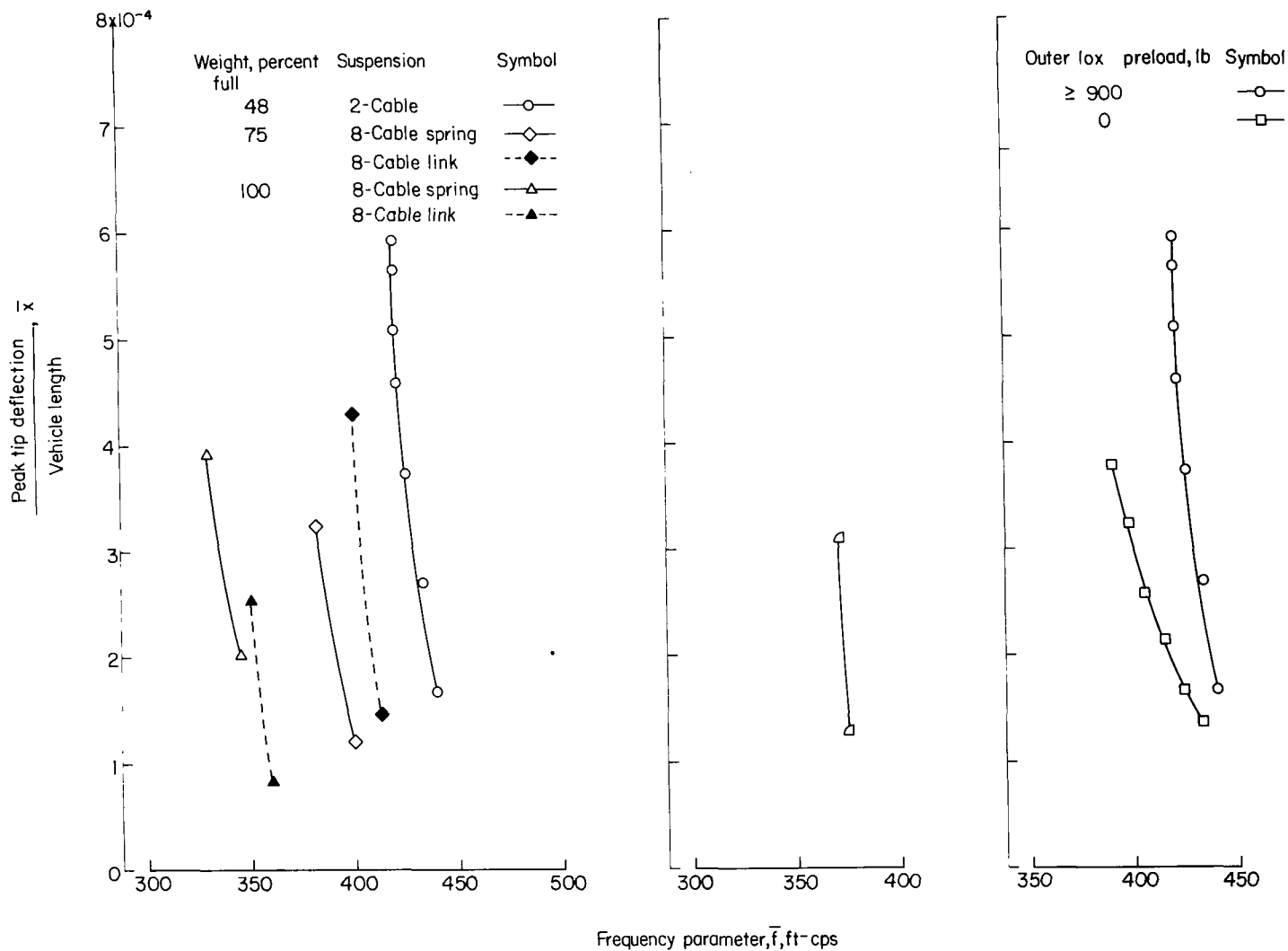


(a) First-bending-mode frequency.



(b) Rigid-body rocking frequency.

Figure 18.- Variation of first-bending-mode frequency and rigid-body-rocking frequency of Saturn with booster water level.



(a) Effect of suspension and weight variations. 1/5-scale model; outer lox preload ≈ 900 lb.

(b) Full-scale vibration test vehicle. Block II(SAD-5). Lift-off weight.

(c) Effect of outer lox tank preload. 1/5-scale model; two-cable suspension; weight at maximum dynamic pressure.

Figure 19.- Variation of resonant frequency of Saturn first bending mode with vibration amplitude.

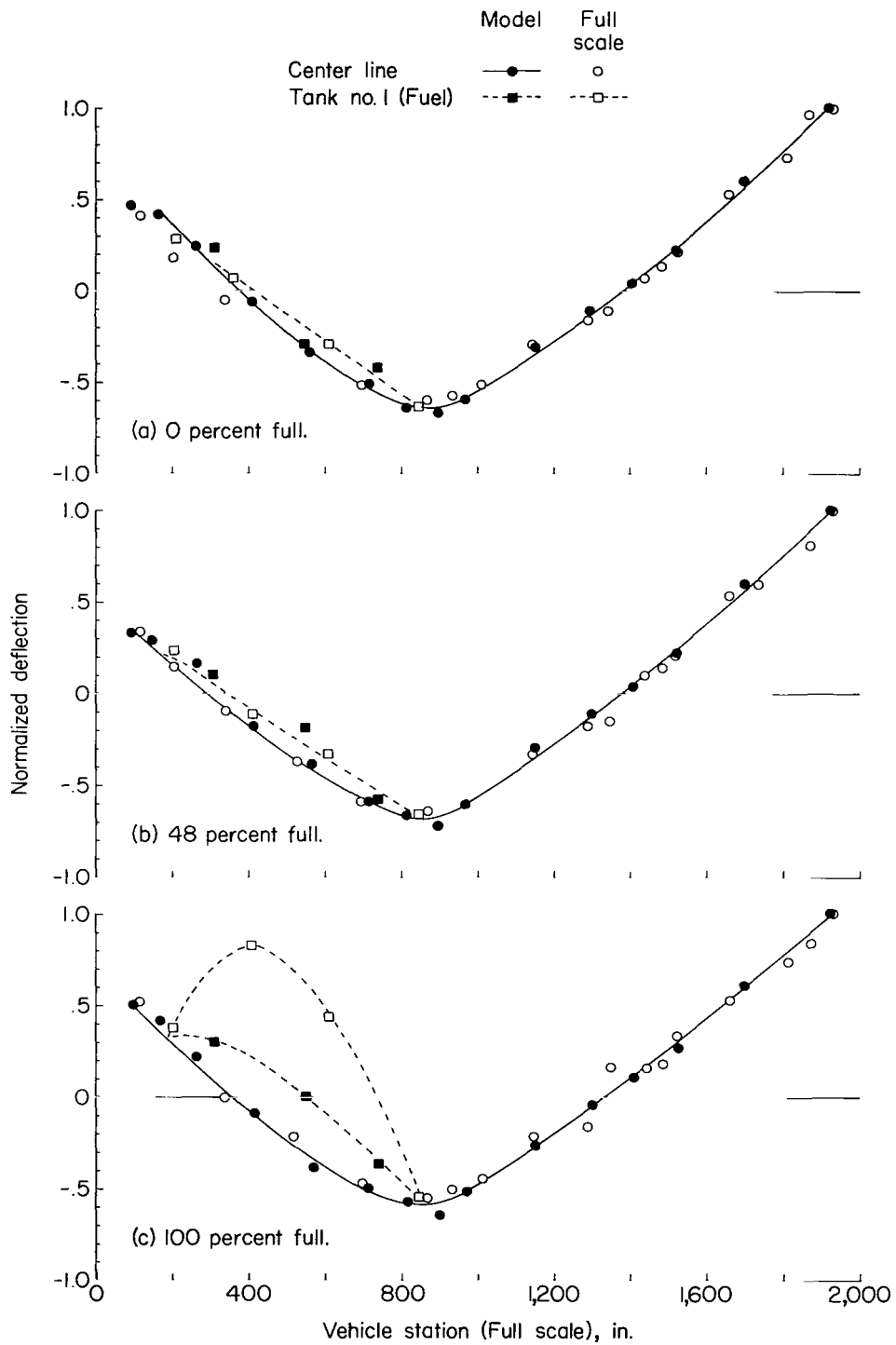


Figure 20.- Comparison of model with full-scale first bending modes.

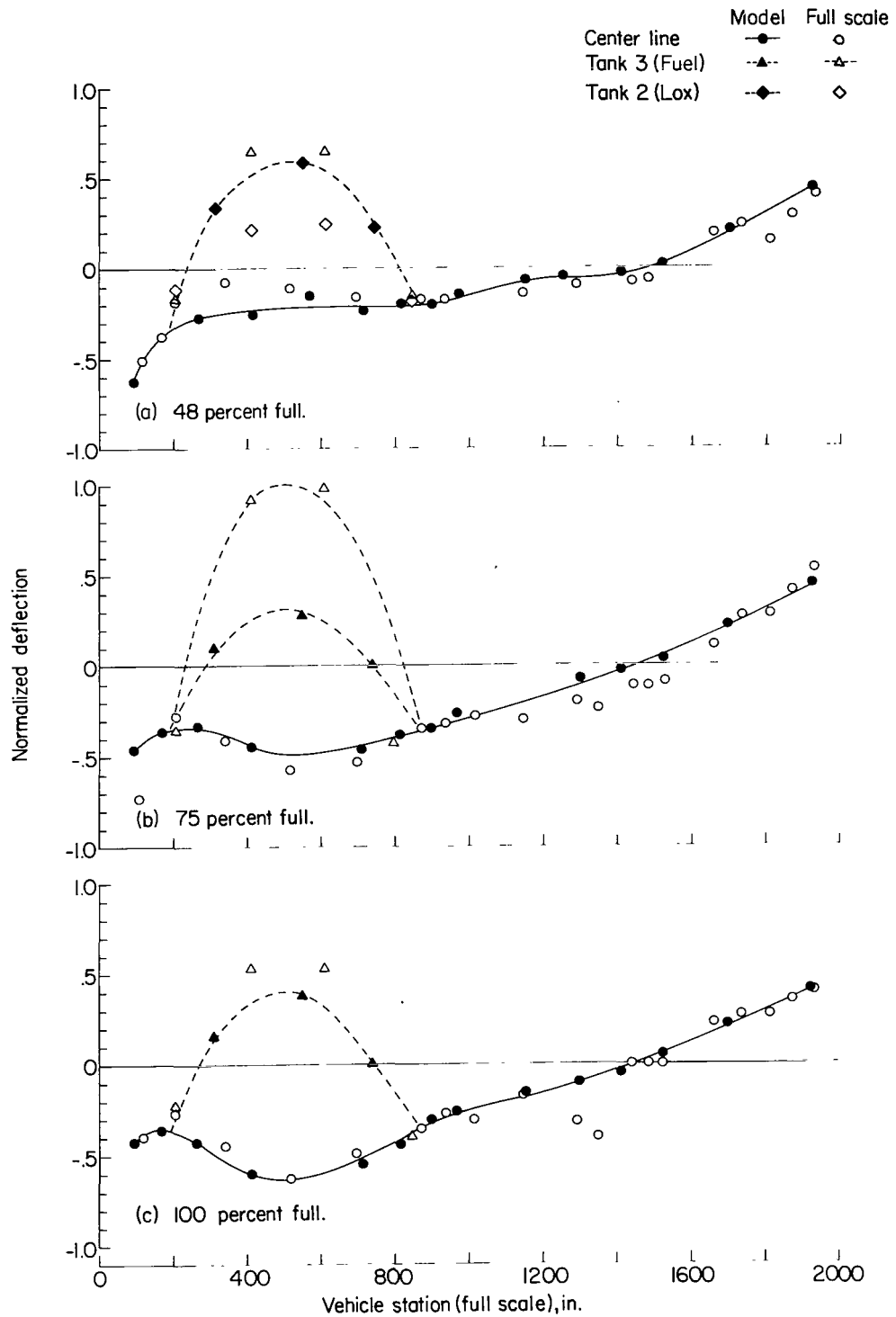


Figure 21.- Comparison of model with full-scale first cluster modes.

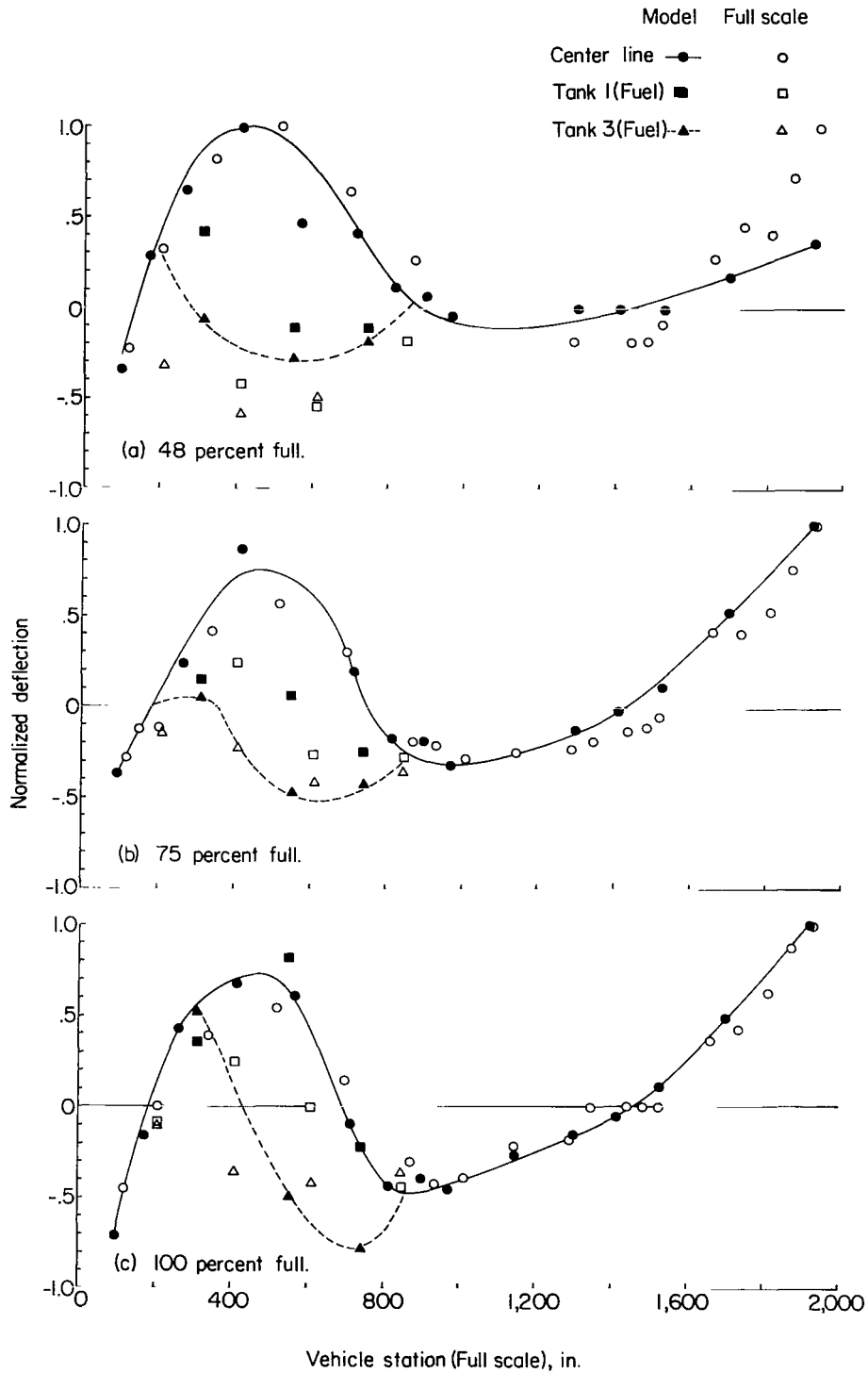


Figure 22.- Comparison of model with full-scale second cluster modes.

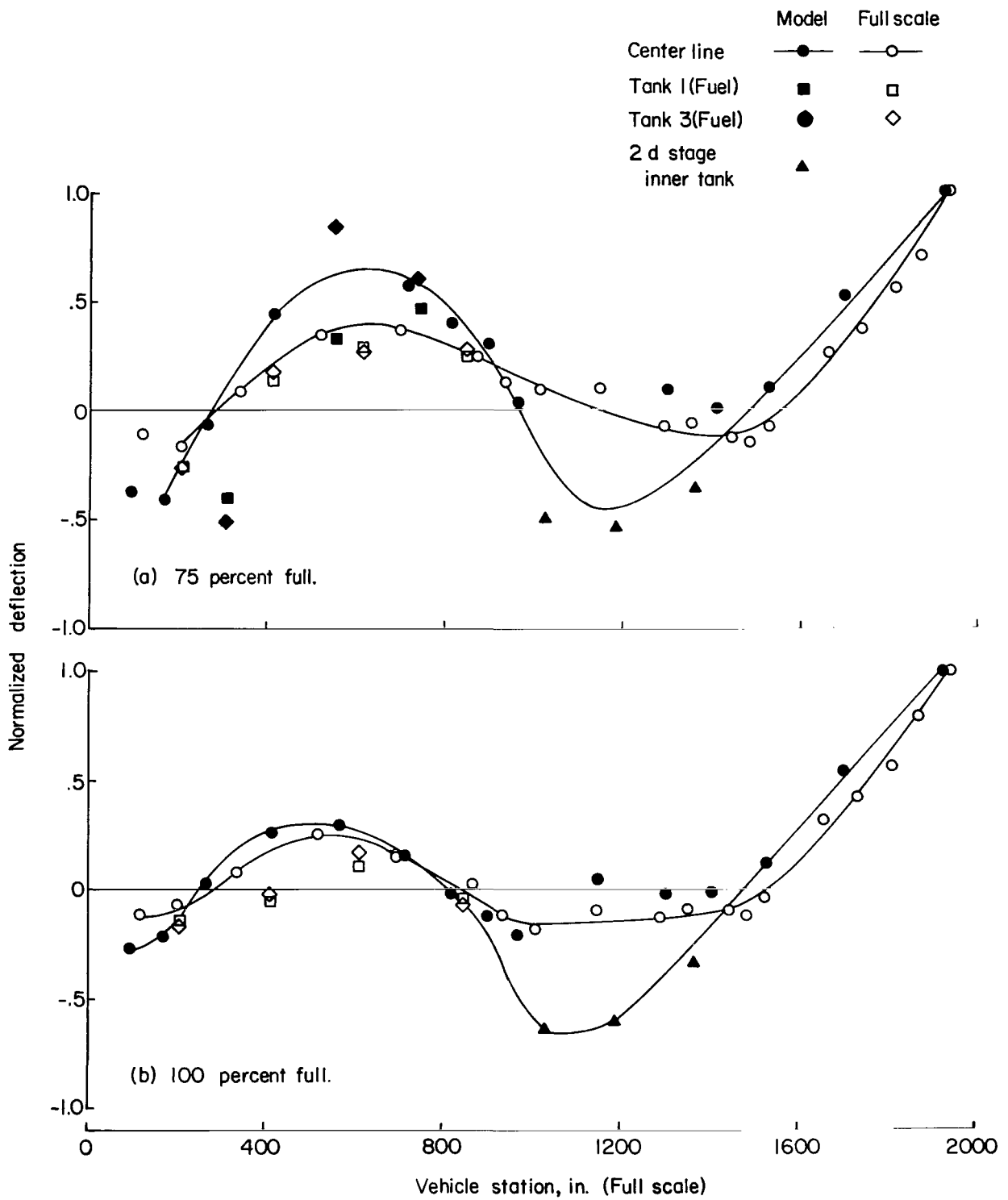


Figure 23.- Comparison of model with full-scale second bending modes.

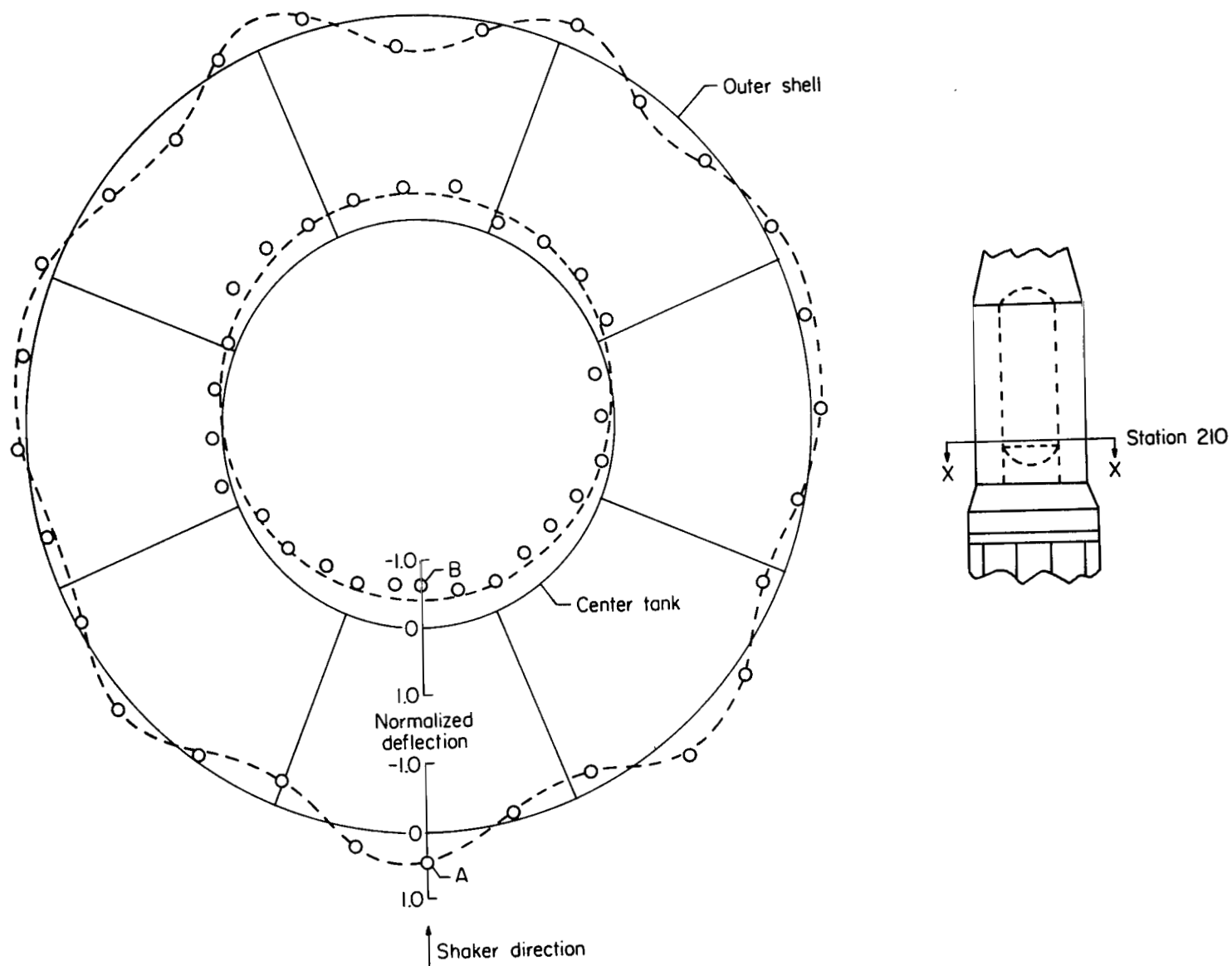


Figure 24.-- Cross-section mode shape at model station 210; second bending mode; frequency, 38.9 cps; booster tanks 48 percent full.

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